Part 1 of 2 January-February, 1955

LUBRICATION ENGINEERING

RESEARCH APPLICATION EDUCATION MAINTENANCE

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Vol. 11, No. 1

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LUBRICATION ENGINEERING

RESEARCH APPLICATION EDUCATION MAINTENANCE

Vol. 11, No. 1 January-February, 1955

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LUBRICATION ENGINEERING is published bi-monthly by the American Society of Lubrication Engineers, 84 E. Randolph St., Chicago 1, Illinois; Phone DEarborn 2-1298. Current single copies 75c; Subscription rates $4.00 per year in United States & Canada, $4.50 in all other countries.

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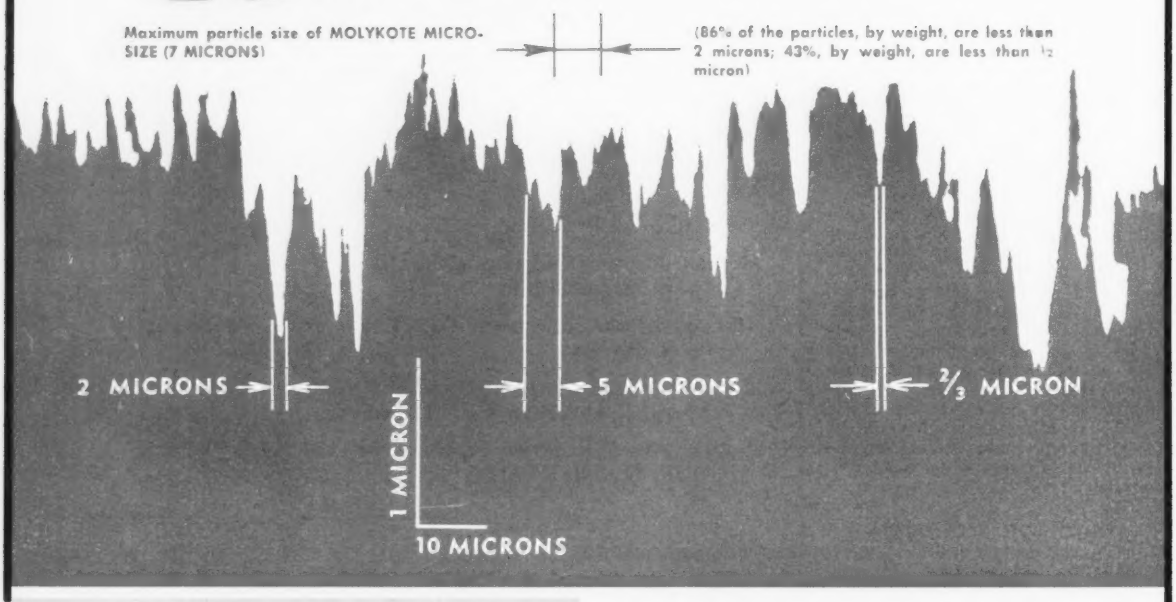
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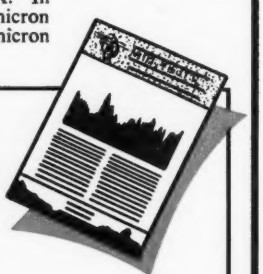

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Note: Information on MOLYKOTE appears in Sweet's Product Design File 1955, Section 3C.

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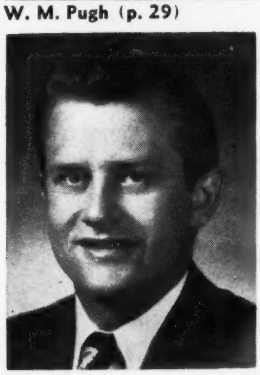

W. Tabor (p. 22)

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L. M. Tichvinsky (p. 29)

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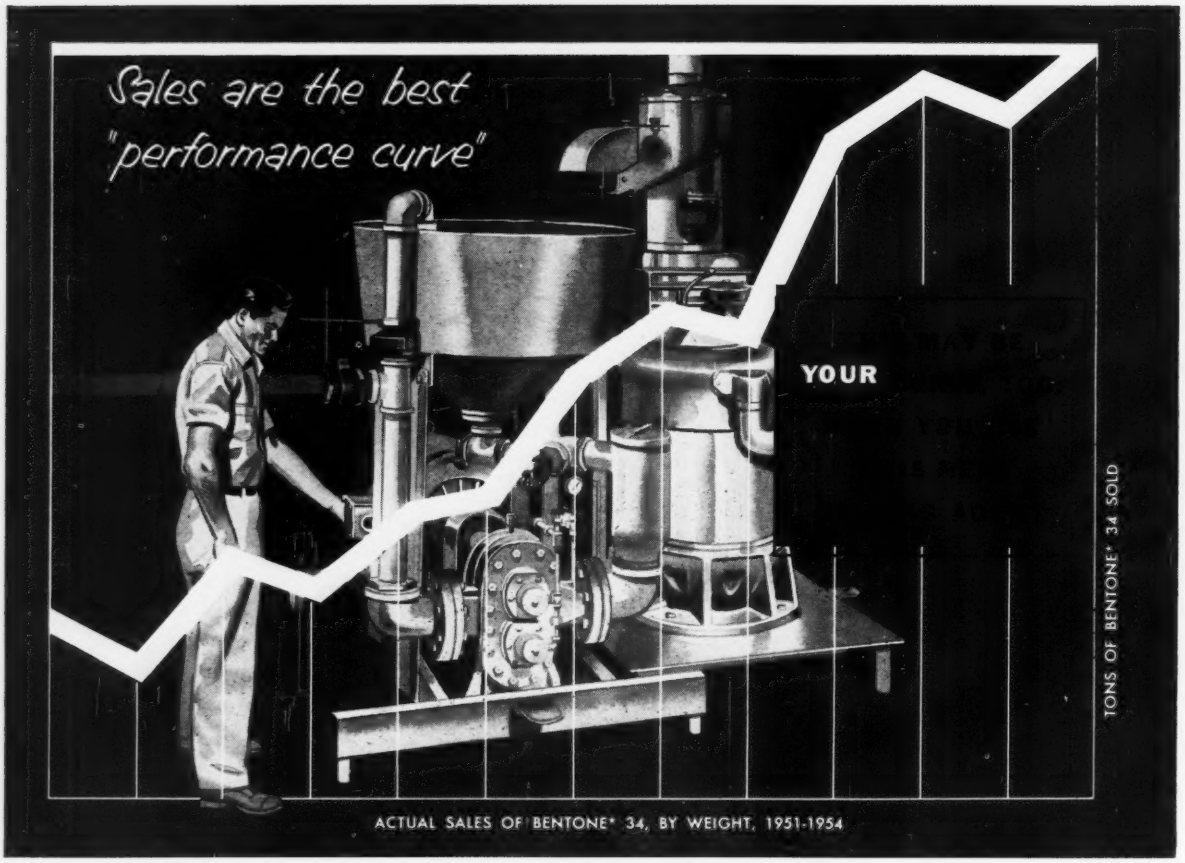

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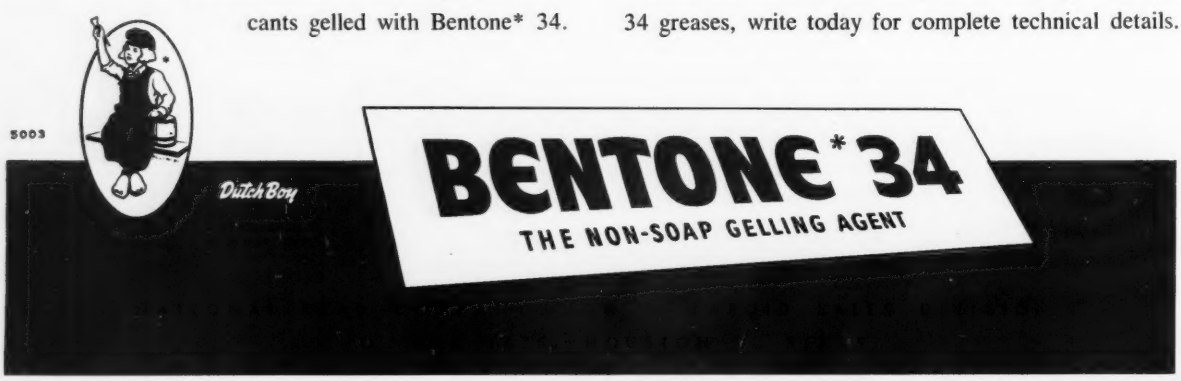
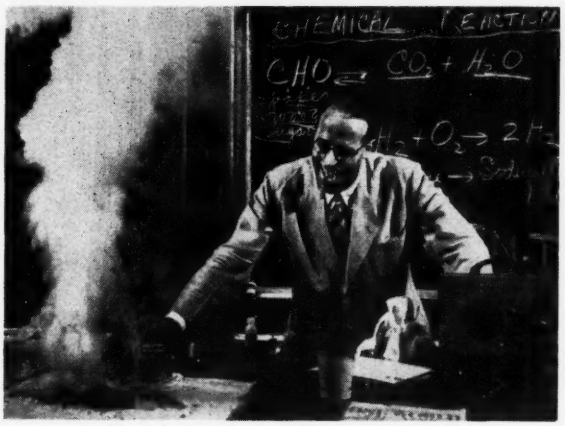


Lube Lines



ANNUAL MEETING BANQUET SPEAKER. Dr. Hubert N. Alyea (pictured above) has been chosen as guest speaker for the ASLE 10th Annual Meeting Banquet to be held on Tuesday evening, April 14, and will present **Atomic Energy: Weapon For Peace** as his address. (Following graduation from Princeton University in 1925, Dr. Alyea spent a year at the Nobel Institute, Stockholm, returning to receive his Doctor's degree from Princeton in 1929. The following year was spent at the University of Minnesota studying the chemical effects of radium, and the next year at the Kaiser Wilhelm Institute in Berlin investigating the explosion of gases. Since then he has been in the chemistry Department of Princeton where his field of research includes plastics, retarding chemical reactions, and the action of light, radium, etc., on these reactions.)

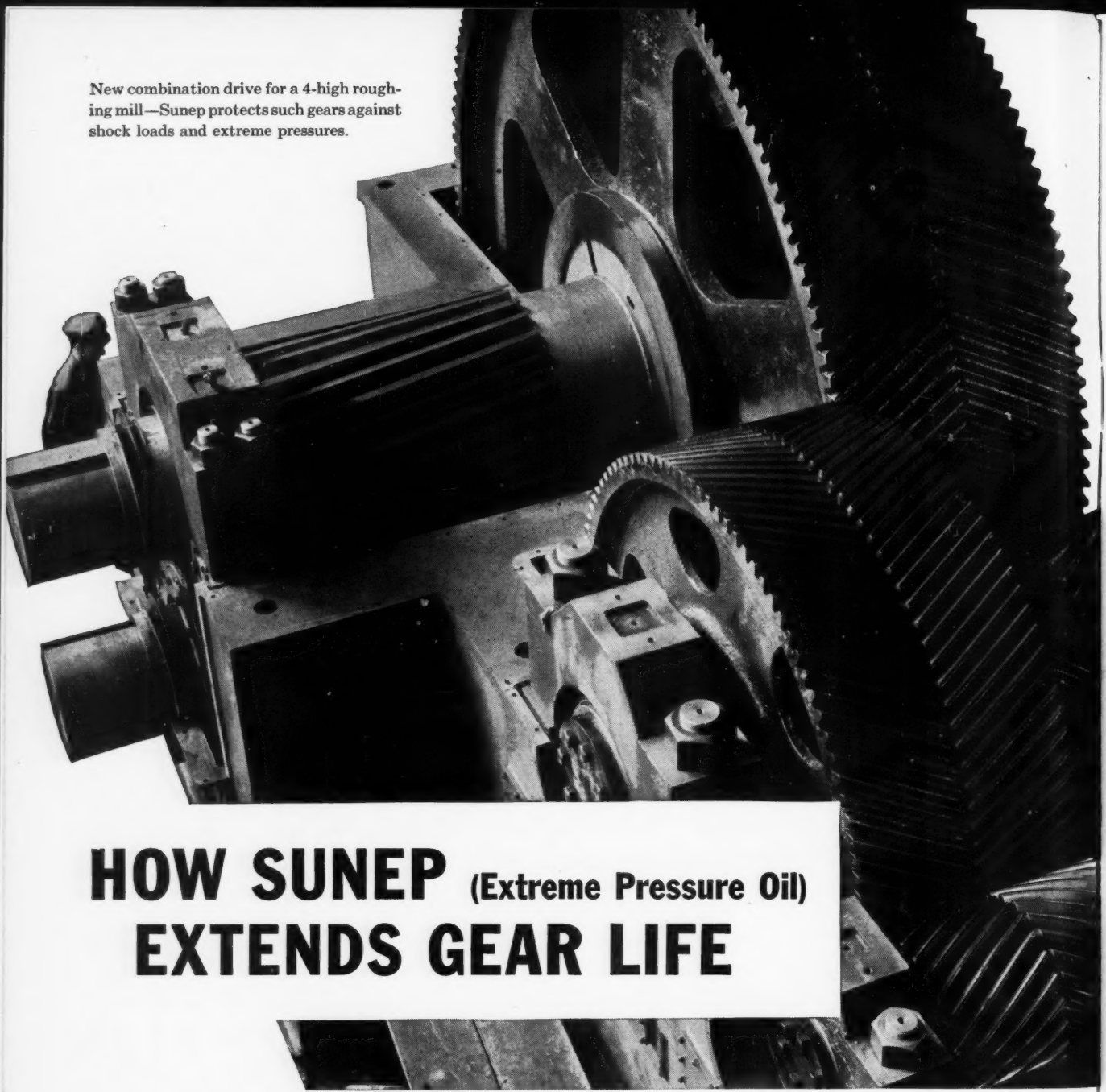
In **Atomic Energy: Weapon For Peace,** Dr. Alyea will trace the growth of ideas which led to the atomic bomb: of the vain attempts of the alchemists of the Middle Ages to transmute lead into gold; of Becquerl's accidental discovery of radioactivity, and of the Curies' isolation of radium; of Rutherford's interpretation of radioactive disintegration and the nuclear atom, and of his achievement of artificial transmutation. How gradually we came to understand atomic structure, and isotopes, and how to smash atoms. He will contrast the actions of ordinary high explosives with those of nuclear fission reactions, and outline the work of the Manhattan Project in exploiting these reactions. The nature of the H-fission bomb will be discussed, and a full treatment of the atomic pile will outline the various ways that new elements are produced in it.

"MEN OF LUBRICATION", by Frank Archibald of Arthur D. Little, Inc. Almost everyone having more than just a casual acquaintance with lubrication problems has heard the name of Osborne Reynolds who was one of the most important contributors to the science. However, very few have had an opportunity to learn anything about the man himself and fewer still have seen his picture. Unfortunately, general knowledge about the other great men in the field is similarly lacking.

Now, in a series of articles commencing with this issue of **Lubrication Engineering,** Frank Archibald presents short biographical sketches of "Men of Lubrication" in which he briefly tells something about their lives and their contributions. In most cases the articles will include a photograph or sketch of the individual. (See page 9 for the first article in this series.)

NEW "LE" COVER. The Editorial Committee, the Board of Directors, and the National Office staff have been hard at work to steadily improve **Lubrication Engineering** and to make it the best source of information on lubrication subjects and the medium in which authors will want to have their work published. The new cover, which appears for the first time with the current issue, is part of the over-all plan, and the changes that have been instituted have been made to improve its appearance and to set forth the purpose of the magazine in a clearer manner. In selecting a new design many points were considered, but the main criteria were that the cover should be simple, attractive, and that it should clearly set forth the title, the name of the Society, and the field of influence of the Journal. The most conspicuous change is the elimination of the oil drop symbol. This was done because the symbol tended to suggest an unintentional limitation to the science of lubrication engineering. The latter is a broad field and embraces all phases of industry and science where lubrication is concerned. Another change is the introduction of the four words "Research, Application, Education," and "Maintenance." These have been added to spell out the fields of influence of Society activities so as to make it clear that **Lubrication Engineering** serves all of them and not just one or two. An innovation, insofar as the Journal is concerned, will be the occasional incorporation of the table of contents on the cover. Over fifty different designs were made before the final cover was selected, and the Society is indebted to several of the industrial members for making their art departments available. It is especially indebted to Messrs. J. Boyd & H. N. Kaufman for their able assistance.

COURSE IN "LUBRICATION ENGINEERING." A two-week summer course in "Lubrication Engineering," co-sponsored by the University of California Engineering Extension and the ASLE Northern California Section, has been scheduled to be held June 20 thru July 1, 1955, on the Berkeley Campus of the University of California. The program will cover: Fundamentals of Lubrication, Petroleum Technology & Properties of Lubricants, Bearing Design & Bearing Materials, Lubrication of Journal Bearings, Corrosion & Its Prevention,



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DECENNIAL SUBJECT INDEX. ASLE is indebted to Messrs. R. K. Gould of The Texas Co., and R. H. Josephson of the Cleveland Graphite Bronze Co., for their classification of the Decennial Subject Index which appears as a supplement to this issue (Part 2 of 2).

WORLD USAGE OF LUBRICANTS. There is a rapidly growing interest in the position of the rest of the world on lubricants in relation to that of the United States. Many estimates have been made and widely scattered statistics have been compiled in order to establish a clear picture. The few comments and figures offered here are without any intent to appear as an authority on the subject, and it must be recognized that there is at least some element of uncertainty in almost all of this type of data. First off, reference to the world outside the United States does not include Russia or her satellite countries. Further, reference will be made to 42 gallon barrels on a calendar day basis.

World consumption of lubricants currently runs perhaps just over 200 thousand barrels per day, which is slightly more than what it was in 1951. U. S. demand is 113 thousand barrels per day or close to 56% of the total. In addition to that which is used domestically, the U. S. petroleum industry has exported to the rest of the world a volume equivalent to about 1/3 of home consumption. This leaves approximately ¼ of world consumption which has been refined outside the United States. Or to put it another way, the rest of the world has been producing slightly in excess of ½ their total needs. This picture, however, is destined to change due to increasing productive capacity outside the U.S.A. as indicated later on in this article.

It is interesting to note that the broad breakdown by oil type indicates that in the U. S. about 1½ times as much automotive lubricating oil is consumed as industrial oil. On the other hand Western Europe, which uses 1/5 of the world's lube oil, consumes about ½ as much automotive oil as industrial oil. The most obvious reason for this difference is the far greater motor vehicle population in the U. S. as compared to the rest of the world. Accurate figures have not been developed for other areas, but it is reasonable to assume that the ratio is even heavier in favor of industrial oils.

Let us take a look at the types of oils which in total have been exported from this country at the rate of around 40 thousand barrels per day during the past year. The ratio of exported automotive to industrial oil follows the same pattern as indicated above for Western European consumption or a ratio of 1 to 2. In the 2/3 portion representing industrial oils, ½ of this total consists of shipments of ordinary red and pale oils. Bright stock and cylinder oils take up 1/7 of the total export. Incidentally, the far greater portion of the bright stock is HVI solvent-refined oil, and Pennsylvania oils still account for the largest share of the cylinder oils.

In the 1/3 portion representing automotive oils, an ever increasing share is of the so-called premium additive-type. While accurate figures are not available, we would guess that the quantity of this type oil must now be approaching 50% of all automotive oil shipped with the possibility it already exceeds this proportion. Of the remainder, there is a good portion represented by high quality oil. Automotive gear oils (excluding hydraulic) are exported at a ratio of around 1 gallon for every 10 gallons of automotive engine oil shipped.

Out of the total of all grades of oils exported, 1/15 goes for non-automotive diesel engine service, and in this field there is a definite preponderance of high additive-type oils with MIL-0-2104 oil accounting for the largest portion. There is, however, still a fair amount of good quality straight-mineral solvent-refined coastal-type oil shipped for medium and slow speed diesel service. The remainder of the oils include various amounts and grades ranging from black oils to medicinal white oils, and embraces insulating oils, cutting oils, etc.

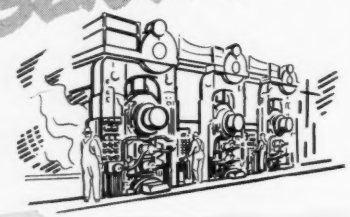
The U. S. also provides the rest of the world a portion of their needs of lubricating greases, exporting over 100 million pounds per year at a rate of 1 pound of grease for every 5 to 5½ gallons of oil.

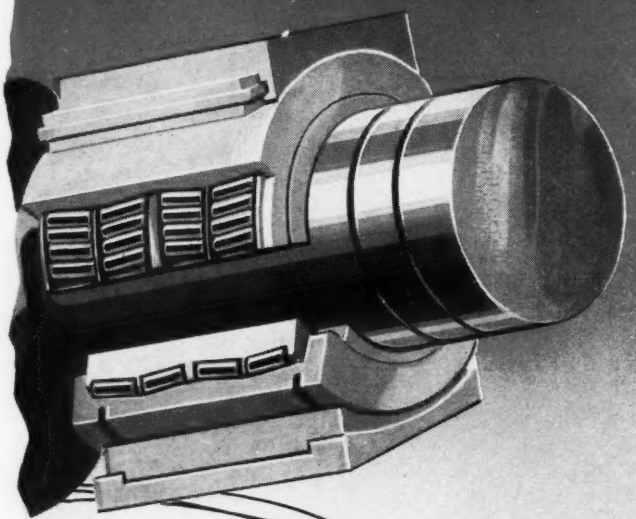
Let us now take a look at lube oil productive capacity. The U. S. Petroleum industry has refining facilities for producing in the neighborhood of 190 thousand barrels per day, which represents 95% of the entire world demand. Just a few years back the productive capacity outside the U. S. was under 40 thousand barrels per day and the best available figures have it that this capacity will have been nearly doubled by 1955. Needless to say, there will be a more than adequate amount of productive facilities, and the excess will necessitate the exercise of wise judgment on the part of the world-wide petroleum industry to maintain production at a level which will guard against a serious surplus inventory of finished lubricants.

As to types of oils, several years ago foreign lube oil production was about equally divided between solvent extraction and conventional refining. In the solvent extraction group some portion has been finished to the equivalent of what is known here as "MVI Solvent Coastal" with the remainder being "Solvent HVI." The rapid technical advances in recent years in the preparation of mineral lubricating oils and the improvements in refining technology have now enabled refiners outside the U. S. to manufacture from Middle East Crude most of the basic lubricating oil stocks which, until a few years ago, were available for the most part only

(Continued on Page 55)

FOR SEVEREST ROLLER BEARING SERVICE





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Men of Lubrication

HISTORICAL INTRODUCTION. While the practice of lubrication goes back to remote antiquity an understanding of its fundamental principles was not forthcoming until relatively recent times, and even today there is much that is not clear under the very common condition of partial lubrication. The researches of the early workers had little influence on the final development of the subject in its rational form and the science of lubrication can be fairly stated to have originated with the work of Beauchamp Tower and Osborne Reynolds. However, some of the earlier work is not without interest and a history of the subject would be incomplete without mention of Amontons, Coulomb, Morin, Hirn and Thurston. The work of N. P. Petroff in Russia deserves special mention.

The year 1883 was an auspicious one for the Science of Lubrication for in that year the work of both Tower and Petroff appeared. Very little is known of Petroff and the difficulty of the Russian language no doubt prevented an early appreciation of the significance of his work by even the more scientific engineers. Tower's work was done under circumstances much more favorable to its spread in engineering and scientific circles, and soon any gain which might have come from an earlier knowledge of Petroff's work was more than made up. The time was ripe and the men were at hand.

Tower's work appeared in a series of four reports to the Institution of Mechanical Engineers. The first two of these reports were on journal bearings, the latter two on collar and pivot bearings; the series extending from 1883 to 1891.

In his experiments on journal bearings Tower discovered that under conditions of copious lubrication a journal and bearing were actually separated from each other by an oil film. This film was found to be under high pressure when measured at various points by an ordinary Bourdon gage. Later it was found that graphical integration of this pressure over the journal area gave a result very close to the actual load on the journal and hence it followed that lubrication under this circumstance is hydrodynamic in nature. Tower also observed a virtual independence of friction torque and load, and the linear dependence of the former on speed. Both these observations confirm the hydrodynamic nature of film lubrication though Tower evidently expected a dependence on the square of the speed from his knowledge of the flow of water through pipes.

The next important step in the history of the theory of lubrication was taken in a very unassuming way but may well have been the result of much reflection. In 1884 the British Association for the Advancement of Science met in Montreal. Lord Rayleigh was president. In his presidential address Lord Rayleigh, in commenting on various scientific advances of the time, mentioned Tower's work, and his words which follow are of sufficient significance to give in some detail:

"As also closely connected with the mechanics of viscous fluids, I must not forget to mention an important series of experiments upon the friction of oiled surfaces recently executed by Mr. Tower for the Institution of Mechanical Engineers. The results go far towards upsetting some ideas hitherto widely admitted . . . In order to maintain its position the fluid must possess a certain degree of viscosity proportionate to the pressure, and even when this condition is satisfied it would appear to be necessary that the layer should be thicker on the ingoing than on the outgoing side. We may, I believe, expect from Professor Stokes a further elucidation of the process involved . . . "

It was, perhaps, no accident that Rayleigh should have had such thoughtful things to say on this subject because he and Tower were personal friends, and it is not at all unlikely that Tower talked to Rayleigh about his interesting results from the friction experiments. Stokes (Sir G. G. Stokes, Lucasian Professor of Mathematics, Cambridge University, 1819-1903) did not respond to Rayleigh's suggestion and it was left to Osborne Reynolds to rationalize Tower's work. Rayleigh was to return to the subject many year later to make an important contribution.

Osborne Reynolds' great paper of 1886 is the real foundation of the Science of Lubrication. Whether Reynolds discovered for himself the basic requirement in the shape of a lubricating film will probably never be known. Without doubt Reynolds was fully capable of arriving at this concept by him-

---

The author of this series, **Frank Archibald** (pictured center), is an expert in the science of lubrication in his own right and has made an avocation of studying the lives and accomplishments of the great men in the field. He was born in Vancouver, Canada, in 1914, but grew up in Harbour Grace, Newfoundland, where his people were in the shoe manufacturing business. After elementary and high school he began engineering study at Memorial University, St. John's, Newfoundland, receiving his degree in Mechanical Engineering in 1936 from McGill University, Montreal, Canada. Coming to the United States in 1938, he was employed by Brown & Sharpe Manufacturing Co., Providence, Rhole Island, in their experimental department until 1946. Following a year of graduate work at Princeton University where he obtained his Master's degree, he taught in the Engineering School, Princeton, from 1947 to 1951. Since 1951 he has been with Arthur D. Little, Inc., in their applied mechanics group, and has contributed to the literature of metal cutting, power chains, theory of lubrication, and machine shop theory.

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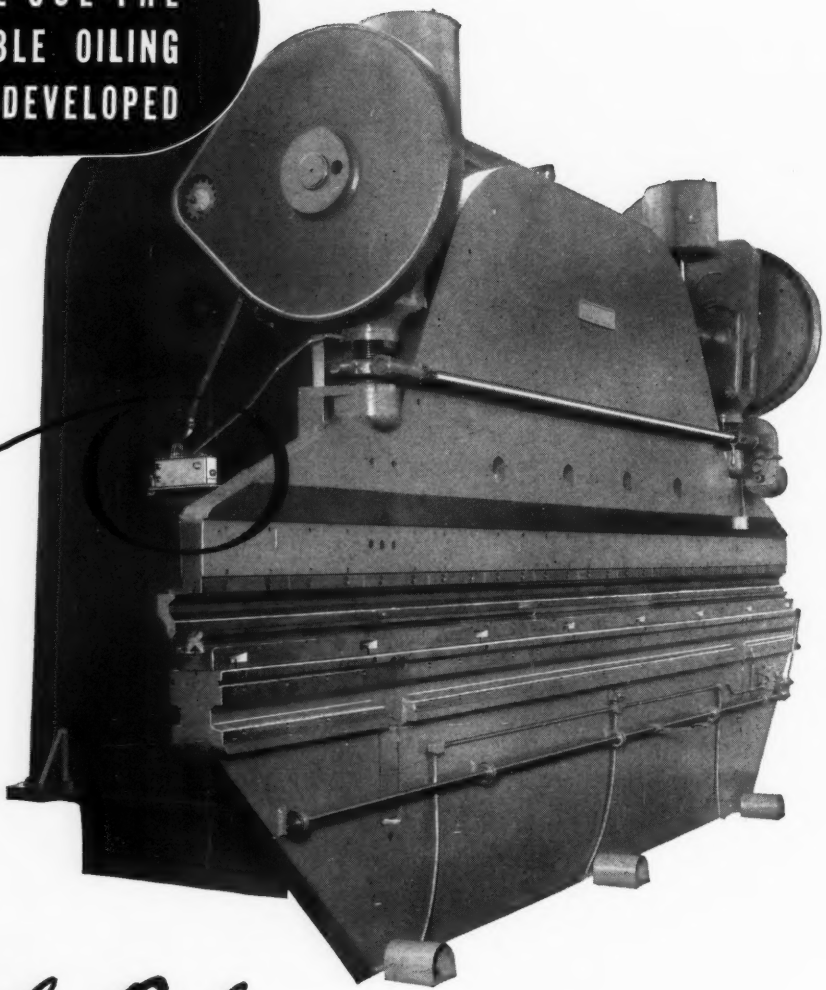

Illustrated is Madison-Kipp Lubricator Model FD installed as original equipment on a ⅜" by 20' Cincinnati Press Brake, manufactured by the Cincinnati Shaper Co., Cincinnati, Ohio.

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self. In any event the paper gives a great deal more than this single idea and is a model of the application of theoretical principles to the interpretation of experimental results.

One of the great disappointments of mathematical physics in the nineteenth century had been the failure of theoretical hydrodynamics to agree with observations on fluid behavior in the way the theory of elasticity had been confirmed by experiments on solid bodies. Reynolds had recently (1883) thrown much light on this difficult matter and had established the well-known criterion demarking what is now called laminar flow from turbulent flow. He saw, from the nature of Tower's results, the probability that the phenomenon of film lubrication fell in the category of laminar flow and hence that the equations of hydrodynamics of viscous fluids would apply. The theory developed by Reynolds has been tested in many ways since that time and is now considered to be in complete accord with observation in all cases where the assumptions are not violated.

It is rather surprising that results of a practical nature did not follow immediately upon this clarification of the subject. Even Reynolds himself seems to have missed the implications of his own work. Actually it may be impossible to gauge completely the immediate effect of Reynolds' paper because of the obscurity of much engineering work. There were at least two men who saw the possibilities of application but their words have attracted little notice.

In the discussion of the Third Report of the Research Committee on Friction of the Institution of Mechanical Engineers, May 1888, Professor Archibald Barr, referring to Reynolds' explanation of film lubrication in a journal, said:

"In an end thrust bearing the case which would correspond to some extent with that of the journal . . . would be, not to have an absolutely flat bearing surface, but to cut one or more radial oil grooves in it and make the surface of the footstep very slightly screw shaped . . . "

The same idea seems to have occurred to a Mr. M. Holroyd Smith (ibid).

But the first serious application of the principles given by Reynolds was made simultaneously by Albert Kingsbury in the United States and A. G. M. Michell in Australia around the turn of the century. Michell's invention of the thrust bearing was accompanied by a solution to the mathematical problem associated with it when the so-called side leakage is considered. Reynolds had neglected this very significant matter in order to obtain his mathematical solutions. Michell's analysis was therefore a very important extension to Reynolds' theory. The mathematical problem solved by Michell in taking account of side leakage is difficult and the computations are so lengthy that only a very few reductions of some special cases have been made. In recent years the problem has been reexamined collaboratively by M. Muskat, F. Morgan and M. W. Meres with some improvement in the results but it remains a formidable mathematical exercise. Kingsbury later contributed to the science of lubrication notably in his electrical analogy (1931).

In the integration of the pressure in a journal bearing Reynolds resorted to the use of trigonometric series. This resulted in very long computational work and worse still the series did not converge for those cases where the journal eccentricity was greater than 0.5. In 1904 Professor Arnold Sommerfeld, who then held the chair of Technical Mechanics at the Technische Hochschule in Aachen, introduced a substitution which makes the integration possible in closed form for all values of the eccentricity. This was an exceedingly valuable addition to the theory of lubrication of journal bearings.

The work of Tower, Reynolds, Kingsbury, Michell and Sommerfeld forms the basis for the whole science of lubrication which is still developing. In the broad sense there was a lull in activity in the subject following their progress which extends from about 1905 to about 1913. In 1913 and again in 1919 W. J. Harrison of Cambridge University made some important contributions to journal bearing theory, and in 1914 an important paper by M. D. Hersey (then of M.I.T.) appeared in which the method of dimensional analysis is used. In 1918 a paper by Lord Rayleigh appeared in which the calculus of variations was used to find an optimum shape for a slider bearing. The result was of considerable practical interest but was not applied until quite recently.

During the 1920's the work of H. A. S. Howarth, S. A. and T. R. McKee in the United States and R. O. Boswall in England is perhaps the most important. The publication by Boswall in 1928 of his book "The Theory of Film Lubrication" is certainly a milestone and the book is one of the finest treatments of the theory there is. Also during this period an important paper by G. I. Taylor on the stability of the fluid between two rotating cylinders appeared. This becomes a matter of importance in high speed bearings, and Taylor's findings as applied to journal bearings have been confirmed in recent years (1950) by D. C. Wilcock. The phenomenon of vibration or shaft whirl in bearings was observed by B. L. Newkirk during this period (1924).

Following Boswall's book in 1928 came Hersey's "Theory of Lubrication" in 1936, and Norton's "Lubrication" (posthumously) in 1942. These books gave a new impetus to the subject and since then a very large number of workers have been attracted to the field.

Parallel to the important work in hydrodynamic lubrication has been the study of the difficult subject of partial or boundary lubrication and the so-called property of oiliness. The work of Hardy, Bowden and Tabor, and Ernst and Merchant has added much to this still puzzling subject. But it seems likely that research workers will be occupied for a long time to come before our knowledge in this field will be comparable to the present state of knowledge in film lubrication.

The history of the subject since the Second World War is rather difficult to write concisely because of the large number of workers and the diverse

(Continued on Page 55)

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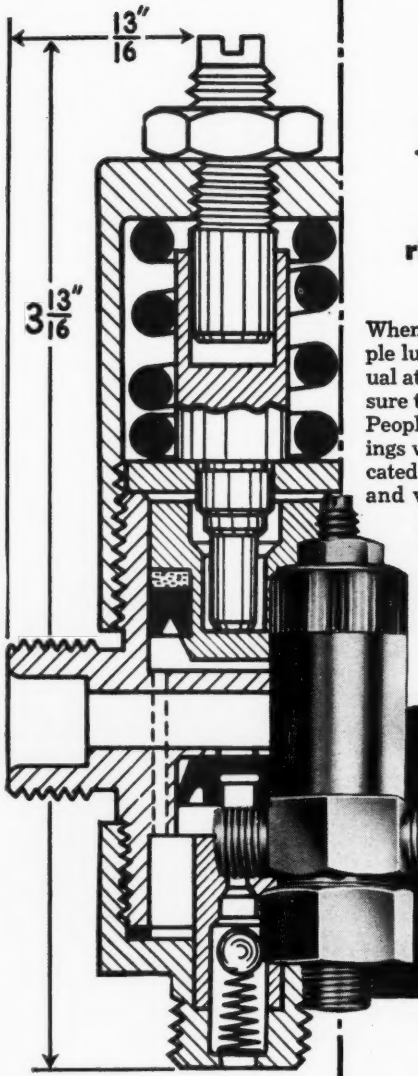

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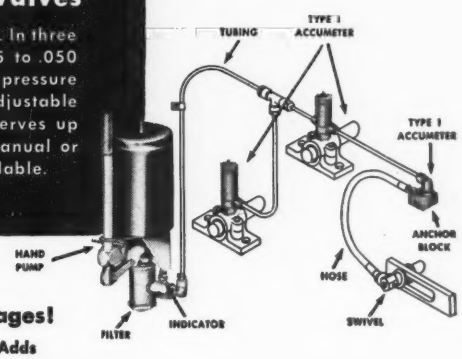

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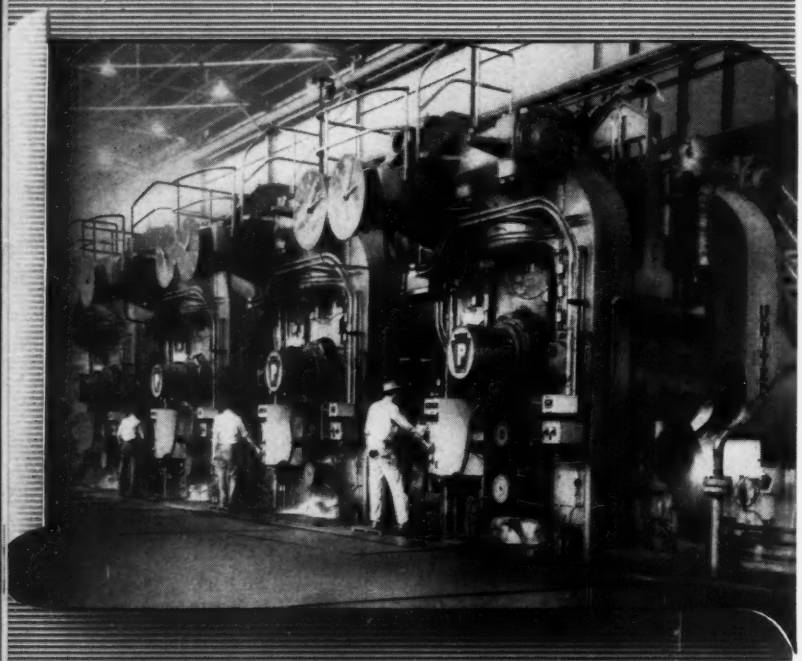

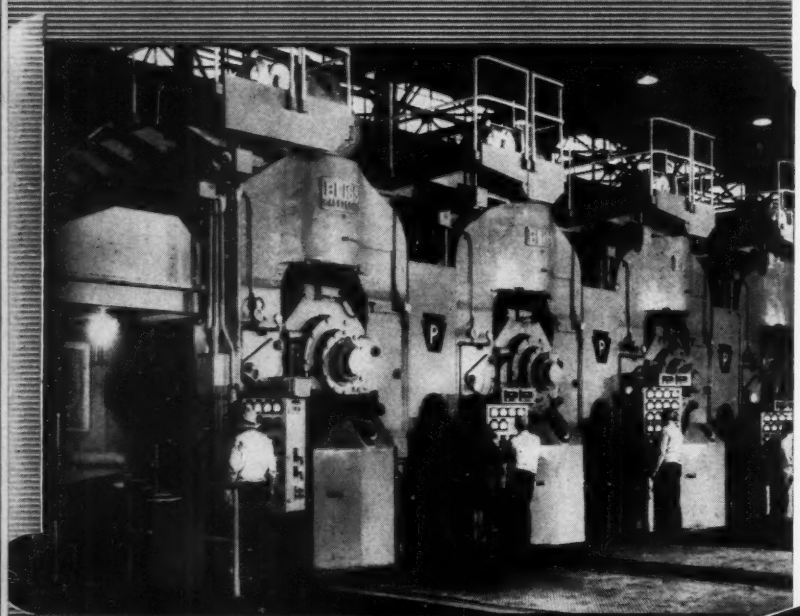

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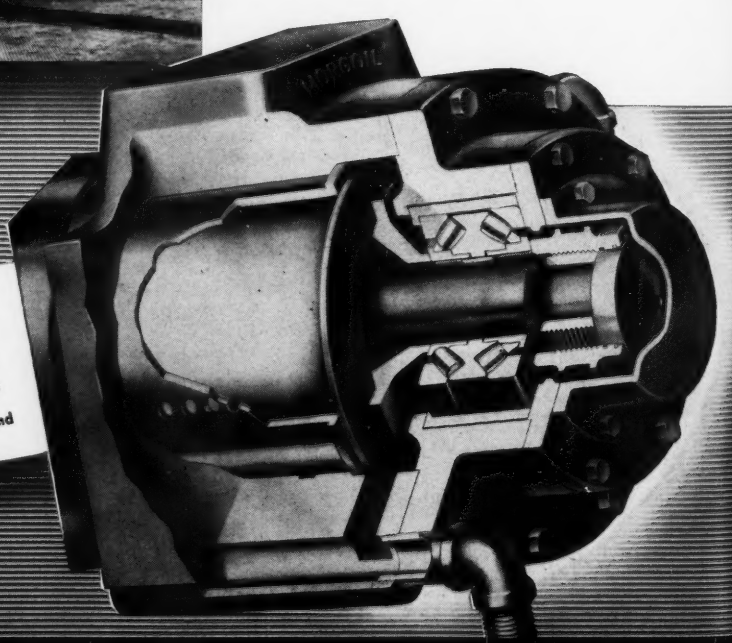

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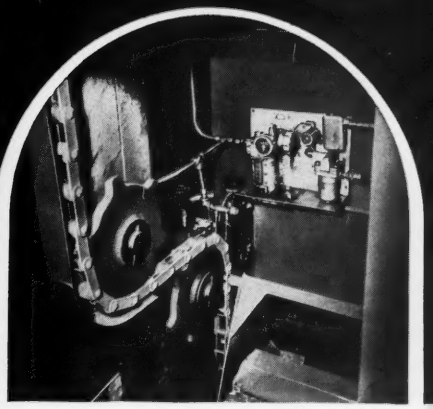

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10th ASLE Annual Meeting & Exhibit

April 13-14-15, 1955 **PROGRAM** Hotel Sherman, Chicago, Ill.

Sixteen technical sessions, a five-session educational course, three panel discussions, a symposium, and a plant tour have been tentatively scheduled as the Program for the 10th Annual Meeting & Lubrication Exhibit of the American Society of Lubrication Engineers to be held in Chicago on April 13, 14 & 15, 1955.

The Panels will include: (1) "Controls for Extending the Useful Life of Coolants While in Use," (2) "Industrial Dermatitis," and (3) "Organization of a Plant Lubrication Program," with D. M. Cleaveland & E. L. H. Bastian, J. W. Hopkinson, and K. S. Smiley as organizing chairmen, respectively.

The Symposium, with A. F. Brewer as organizing chairman, will cover "Plant Safety as Related to Lubrication."

Technical Sessions will be devoted to: (1) Film-Type Bearings, (2) General Industry, (3) Greases, (4) Hydraulic Fluids, (5) Lubricant Dispensing Equipment & Application, (6) Measuring Properties of Lubricants, (7) Metal Working, (8) Modern Concepts of Wear, (9) Present Thinking in Gear Lubrication, (10) Rolling Contact Bearings, and (11) Timely Studies in Friction. Tentative papers include:

Adhesive Wear of Metals, by C. D. Strang

Atoms Trace the Wear, by W. R. Miller & H. R. Jackson

Centralized Grease Lubrication Equipment, by L. T. McDonald

Chemical Compositions of Lubricating Oils, Cutting Oils & Petroleum Solvents, by W. C. Witham

Control of Quality of Water Soluble Machining Coolants During Use, by J. A. Prevel

Control of the Quality & Life of Grinding Coolants During Use, by O. L. Maag

Corrosion Inhibited Greases, by G. H. Link

Critical Survey of Industrial Lubrication Engineering in Great Britain and the U. S. A., by G. D. Jordan

Economic Factors of Centralized Lubrication, by R. Lohbauer

Effect of Pressure & Molecular Weight Upon the Viscosity of Polybutens, The, by L. B. Sargent, Jr.

Effect of Sliding Velocity & Temperature on Wear & Friction of Several Materials, The, by R. L. Johnson, M. A. Swikert & E. E. Bisson

Electron Microscope Study of Lubrication & Wear, by S. B. Twiss, C. R. Lewis & D. M. Teague

Evaluation of Lubricating Grease Compatibility, by A. L. McClellan & S. R. Calish

Factors Influencing Friction & Wear with Solid Lubricants, by M. B. Peterson & R. L. Johnson

Frictional Characteristics of Plastics, by W. C. Milz & L. P. Sargent, Jr.

Future Trends in Sealed Ball Bearings, by L. D. Cobb

Gear Lubrication in the Steel Industry, by J. P. Critchlow

Heat Treating & Quenching Oils, by R. Weir

High Speed Friction & Wear Machine, A, by H. G. Clarke, Jr. & W. W. Shugarts, Jr.

Influence of Moisture on the Friction & Surface Damage of Clean Metals, The, by R. O. Daniels & A. C. West

Laboratory Appraisal of Drawing Compounds, by W. J. Wojtowicz

Low Temperature Operation of Aircraft Accessories, by E. A. Bancak & R. S. Barnett

Lubricants Dispensing Equipment in Metal Working Machinery Operations, by M. A. Cavanaugh

Lubrication of Rolling Contact Bearings at Elevated Temperatures with Solid, Liquid & Gaseous Lubricants, by Z. N. Nemeth & W. J. Anderson

Maintenance of Quality in Aluminum Sheet Rolling Oils, by J. O. McLean

Mass & Energy Balance for the Wear Process, A, by P. F. Chenea & A. E. Roach

Modern Water Based Cutting Fluids, by A. Manteuffel

Nature & Performance of Some Inorganic Base Greases, by W. H. Peterson & A. Bondi

Neutrons, Gamma Rays & Wear, by A. Hundere, J. P. O'Meara & G. C. Lawrason

New Bearing Metal Theory in Which Lubrication is an Important Factor, A, by B. Lunn

New Measurements of Attitude & Eccentricity in Journal Bearings, by B. Sternlicht & D. D. Fuller

Non-Flammable Hydraulic Fluids (two papers), (1) by W. H. Millett, (2) by C. R. Schmitt

Progress Report on Silicone Lubricants, A, by H. F. Lamoreaux

Pumpability of Aircraft Turbine Lubricants at Low Temperatures, by J. H. Bridges

Radioactive Study of the Metal Cutting Process, A, by I. Finnie & E. Rabinowicz

Radioactive Wear Studies, by A. Hundere

Safety Aspects of Centralized Lubrication, The, by T. G. Carico

Some Unusual Conditions Encountered in the Lubrication of Rolling Contact Bearings, by T. W. Morrison

Study of Fretting Wear in Mineral Oil, A, by D. Godfrey

Study of the Sliding Process of Metals, A, by L. F. Coffin

Survey of Wear Mechanisms, by J. T. Burwell, Jr.

Tool Weight Loss for Cutting Fluid Evaluation, by J. P. G. Beiswanger, J. P. Copes & R. L. Mayhew

Use of Glass as a Lubricant in Extruding Steel, The, by J. Strauss

Wear of Iron in Sliding Contact with Various Elemental Metals, The, by C. L. Goodzeit, R. P. Hunnicutt & A. E. Roach

Wear Studies with Radioactive Gears, by V. N. Borsoff

Worm Gear — Operation & Lubrication, The, by S. D. Craine & R. B. Moir

X-Ray Methods Applied by Automotive & Petroleum Research, by F. W. Lamb

RECENT ADVANCES IN THE LUBRICATION FIELD

by B. B. Farrington*

Lubrication is a very broad subject and must be examined from many different viewponts to be understood. The following review is an especially illuminating assessment of recent advances from a chemist's point of view.

The applications of chemistry to the field of lubrication are many and varied. Not only does the petroleum chemist require a thorough knowledge of the physical and chemical properties of petroleum and its fractions, but also he must know much of the chemistry of the numerous additives used to enhance and extend the basic properties of petroleum products, as well as the chemistry of the metal and plastic surfaces and parts with which petroleum products come in contact.

This is especially true of the lubrication chemist who must know the chemistry and physics of petroleum lubricating oils; of oxidation inhibitors; of gum, sludge and wax dispersants; of the theory of lubrication, friction, and wear; and of the chemistry of new bearing materials. He also must see that his lubricants meet the requirements of the jet age with its ever-increasing temperatures and pressures. The applications of chemistry in the lubrication field have increased rapidly in the last ten years, and some of the most recent advances are described in this paper.

PHYSICAL & CHEMICAL PROPERTIES. Molecular volumes of hydrocarbons are used by lubrication chemists to calculate viscosities and other physical properties. Data in the high molecular weight range are of especial interest. Kurtz and Sankin[12] have extended previous calculations of molecular volume at 20 C., and 1 atm., to temperatures from —253 C. to 100 C., and from 1 to 9,673 atm. (10,000 kg/cm^2) in the range 0 C. to 100 C. The equation correlates molecular volume with chemical constitution and has the following form:

$$\text{Mol. Vol.} = F_1K_1N_1 + F_1K_2N_2 + F_1K_3N_3 - F_1K_4N_4 + F_2K_5$$

where N_1 = number of chain carbon atoms; N_2 = number of carbon atoms in rings; N_3 = number of carbon atoms at ring junctions; N_4 = number of double bonds; K_1 to K_5 are corresponding volume increments which vary with temperature; F_1, F_2 are corresponding volume increments which vary with temperature and pressure.

*California Research Corp., Richmond, Calif.

The molecular volume of saturated hydrocarbons can be calculated to an accuracy of 1% or better over a wide range of temperatures and pressures and is said to fit with the concept that the molecular increment in volume is related to the kinetic energy associated with motion of the whole molecule.

Although the viscosities of simple chemical compounds, such as decalin, increase only nominally (1.3 to 7 times) up to a presure of 2000 atm., lubricating oils increase in viscosity several hundred to several thousand times, according to Kuss[13] of the Institut für Erdölforschung, Hanover. The effect of sulfur dioxide treatment of a lubricating oil distillate was found to have a marked effect on the viscosity-pressure curve, the extract (containing aromatics) increasing in viscosity with pressure much more rapidly than the refined white oil. Furfural treatment produced a raffinate having a lower pressure coefficient of viscosity than Duosol (phenol-cresol-propane). Additives of the oxidation-inhibitor and heavy-duty detergent types had little effect on the viscosity increase with pressure of the base lubricating oil.

In cooperation with Dr. A. Bartel of the Institut, Kuss[13] made micrographs of the effect of surface quality of metals by sudden release from high pressures. Polished steel plates were subjected for 24 hours to oil pressures of 10,000 atm., after which the pressure was suddenly released. Considerable surface damage resulted. This effect is believed to occur to some extent in gear lubrication because of rapid changes in loading. The subject of physical properties of lubricating oils at high pressures should not be concluded without mention of the monumental compilation of data on more than forty lubricating oils of diverse types for the Viscosity-Pressure Project of the American Society of Mechanical Engineers[1] carried out at Harvard University. Oils were subjected to pressures up to 150,000 lb./sq. in. (10,207 atm.) and temperatures up to 425 F. (218 C.). Although only a few conclusions have been drawn to date, a wealth of accurate viscosity and density data is available for study. The suspected effect of 1% of an extreme pressure additive, tri-o-cresyl phosphate, on viscosity at high pressure was not found.

Of importance in the separation and analysis of hydrocarbons is adduction by urea and thiourea. Although it has been known that the principal requirement for hydrocarbon adduction by urea is a

long, unbranched chain, a systematic study of thiourea was lacking. Schiessler and Flitter[26] recently have made a comprehensive study of thiourea adduction, with the aid of high molecular weight hydrocarbons synthesized for American Petroleum Institute Research Project 42. It was found that hydrocarbons differing widely in structure form crystalline complexes with thiourea at room temperature if the cross-sectional dimensions of the molecule are 5.8± 0.5 by 6.8±0.3 Ångstroms as estimated from measurement of Fisher-Hirschfelder models. Compounds near the tolerance limits form unstable adducts.

LUBRICATING OILS. There is general agreement that organic hydroperoxides are the critical intermediates in many oxidation reactions of organic compounds, including petroleum lubricating oils. Most oxidation theories explain the pronounced effect of hydrocarbon structure upon ease of oxidation by assuming a significant effect of chemical structure upon the thermal dissociation of the intermediate hydroperoxides. Thomas[30], in a paper presented to the Petroleum Chemistry Section of the American Chemical Society, describes the thermal decomposition of primary, secondary, and tertiary hydroperoxides in medicinal white oil solvents with 0.02 molar concentration of phenyl alpha-naphthylamine, a chain reaction inhibitor. In all cases, first order decomposition kinetics were observed. It is believed that the measured activation energy corresponds to the the true dissociation energy of the peroxide (0-0) bond under the solvent conditions employed; the selected value was 29.0 kcal/mole.

The contamination of internal combustion engine oils with oxidation products of the fuel and of the lubricating oil itself was the subject of study by Evans and Matthews[6]. These investigators examined used motor oils by means of the light microscope and sedimentation techniques and found that the individual carbonaceous particles forming the major part of the solid phase fell within a narrow size distribution (around one micron). Most of the carbonaceous particles were considered to have originated in the combustion zone.

The success with which the electron microscope has been used for studying highly dispersed solid materials has been extended to the field of lubricating oils. Among the first to report on the degree of dispersion of heavy-duty, detergent-type additives was Broughten[3], who obtained electron micrographs of detergents in lubricating oils of various types, as well as of used oils from a Diesel locomotive showing increased agglomeration of carbonaceous material as the compounding material became exhausted. McBrian and Atchison[15] have used the electron microscope to examine used oils from Diesel locomotives and have presented micrographs of various lubricating oil-additive combinations, together with a report of their performance in service.

Peri[22] has made electron microscope studies of detergent type motor oils in which the dispersed particles were well under 300 Ångstrom units in diameter during the period of maximum detergent efficiency. Significant changes in dispersancy of the oil with use in the engine were detected with the electron microscope long before they were apparent by other methods commonly in use. Samples of used oil (originally containing approximately 1% detergent by weight) from single-cylinder, Caterpillar Diesel test engines were examined by means of the electron microscope at intervals up to 160 hours, and the state of particle dispersion compared with the blackening of piston parts. After 40 hours under the test conditions, small, single particles in the used oil displayed a distinct tendency to cluster, forming small floccules of two to ten primary particles. Deposits began to form on the piston at 160 hours, and the used oil at this time closely resembled a used, uncompounded mineral oil at 40 hours. The oil was almost free of colloidally-dispersed particles, and the agglomerates could be removed to a considerable extent by filtration. While the use of the electron microscope for routine examination of lubricating oils in service is not feasible in all cases, the method does provide a means of studying the life of additives in motor oils, thereby expediting their improvement. The electron microscope method also may be of value in large-scale operations in the transportation field.

Peri[23] has obtained for the first time electron micrographs of certain detergent and soap micelles in oil solution. Calcium cetyl phosphate in a naphthenic-type oil showed amorphous masses averaging several microns in diameter but which, at very high magnification, proved to be composed of approximately spherical micelles each having a diameter of about 50 Ångstroms packed into small crystalline regions. Dispersed into the solution, these micelles often formed short, rod-shaped particles, consisting of two or more simple micelles. Micelles in a naphthenic mineral oil of a mixture of calcium cetyl phosphate and sulfurized calcium alkyl phenate proved to be 40 to 50 Ångstroms in diameter, somewhat similar to those of the calcium cetyl phosphate but with a tendency to form into rows in two-dimensional, lamellar paracrystals. Lead oleate in medicinal white oil formed a gelatinous suspension containing nearly spherical particles averaging about 40 Ångstroms in diameter. A tendency to form short chains also was noted in this case.

A distinct advance in automotive lubrication was made recently by the introduction of all-year, multigrade lubricating oils. These lubricants have the good oil consumption characteristics of an SAE 30 grade oil combined with the ease of cranking of an SAE 10W grade product. Manufactured from a selected, narrow-boiling range, petroleum lubricating oil fraction thickened with a polymeric compound, these oils save gasoline in start-and-stop driving because of reduced friction at the high shear rates characteristic of moving engine parts. Frazier, Klingel and Tupa[8] reported that in extended field tests no significant differences in oil consumption could be found between motor oils containing viscosity index improver polymers and straight mineral oils, other than those attributable to variations in their viscosities as measured at 300 F. (149 C.). Highly significant gasoline mileage differences in

favor of the polymer-containing oils were observed. Our own experience has shown up to 15% saving in gasoline mileage.

According to Murphy, O'Rear, and Zisman[20], aliphatic diesters are in increasing demand for extreme temperature range lubrication of turbojet engines, instruments and machine guns and for aviation and ordnance greases. The dibasic acids from which these esters are made are also in great demand for fibers and plastics, and products obtainable from native raw materials are especially desirable. These investigators found that pinic acid derivable from alpha-pinene, a major constituent of turpentine, is a suitable material for the production of ester lubricants if oxidation inhibitors such as phenothiazine are used. Of the pinates, dihexyl and diheptyl pinates appear to offer most promise as lubricants and grease bases.

Zisman[4] and coworkers, in a reexamination of glutarate diesters, found them to be entirely satisfactory for lubricant applications provided certain impurities were removed. This is considered to be of importance because large quantities of glutaric acid may be expected to be commercially available in the near future.

In the search for high temperature hydraulic fluids for aircraft, Mosteller[19] has found that the polyalkylene glycols and their derivatives are more satisfactory than petroleum oils in viscosity-volatility characteristics and viscosity-temperature relationships but require suitable thickening agents to meet the needs of high temperature hydraulic fluids. Silicate esters, such as tetra-2-ethylhexoxy silane and tetra-2-ethylbutoxy silane far exceed petroleum oils in viscosity-temperature and viscosity-volatility characteristics, in some cases approaching the polysiloxanes.

LUBRICATING GREASES. Although the electron microscope has proved to be an invaluable tool in studying the size and shape of the particles of metal soaps commonly used as thickening agents for lubricating greases, systematic studies of the effect of the metal and acidic portions of the soap and of cooling rate on particle dimensions have only recently been made. Hotten[10] obtained electron micrographs of particles from a series of lithium soap greases in which various anions were used. Lithium soaps of saturated carboxylic acids containing 14-18 carbon atoms per molecule formed smooth greases containing fine, rod-like particles, while lithium laurate formed a coarse, mushy dispersion containing larger particles. Lithium stearate formed very fine particles in a naphthenic oil, coarser particles in a paraffinic oil, and still coarser particles in an ester oil. As could be expected, particle size increased as cooling rate decreased. Gelling power of these soaps was found to depend upon particle surface area per unit volume. A more difficult phase of grease structure research is the determination of the over-all structure of the oil-soap system and the effect of the strength of this structure on bulk properties, such as yield point, viscosity, resistance to shearing forces, temperature, etc. Because of their plastic nature, lubricating greases ordinarily cannot be cut into thin slices for examination. Even very thin smears are too thick for electron microscope examination. R. D. Vold, Coffer, and Baker[31] were able to overcome this handicap by shearing off extremely thin (0.05 to 0.1 micron) slices of refrigerated, calcium hydrated stearate-cetane gel, evaporating off the cetane and examining the soap structure by means of the electron microscope. Although the gel structure found differed considerably from that observed by others in conventional calcium-tallow soap petroleum oil greases, this might be expected from the difference in materials worked with. It is believed, however, that much can be done with the microtome-sectioning technique.

M. J. Vold[32] has attacked the primary grease structure problem from the standpoint of flow. A study by X-ray diffraction of the degree of orientation of the grease particles in a rubbed grease surface gave indications that, in the grease and soap pastes studied, grease flow takes place partly as an articulated, complete structure and partly as unit, orientable particles. Elersich[5] has continued this work, with the conclusion that the relative importance to structure of ultimate particles and particle aggregate varies widely from grease to grease.

By extracting the oil from lubricating greases with butylene followed by ethylene, and evaporation of the latter above its critical pressure, sodium soap aerogels were obtained by Peterson and Bondi[24] with loss of only about 2% of the original volume. The grease was then reconstructed by reintroducing the same oil, with close duplication of the original properties. Reconstitution with other oils showed grease consistency to depend upon viscosity and freedom of the oil from adsorbable, polar bodies.

Copper phthalocyanine was reported by Fitzsimmons, Merker, and Singleterry[7] to be an effective gelling agent for a wide variety of petroleum and synthetic lubricating fluids. Greases prepared with this compound retained a useful structure at 150 C. for extended intervals. The aryl ureas were also found by Swakon, Brannen, and Brunstrum[29] to thicken methyl-phenylpolysiloxane (DC-550 oil) to greases having remarkable thermal stability. These products lubricated ball bearing electric motors for periods up to one month at 450 F. (232 C.) and for six months at 325 F. (163 C.); wear properties, however, have still to be improved.

FRICTION & WEAR. Although friction usually is considered to be a predominantly physical property and wear is regarded as fundamentally mechanical, formation of chemical films modify both to a large degree, and chemists are dealing continually with chemical aspects of friction and wear.

West[33] has obtained micrographs demonstrating adhesion of clean, polished surfaces of cold-rolled, low-carbon steel. These surfaces were merely placed together and then sectioned through the region of contact. Breaking of these metal bridges is reflected in friction and wear.

Fretting wear and corrosion, a type of damage characteristic of vibration or small amplitude, oscillatory motion, has been studied by Mason and

White [14]. They found that fretting wear, which is a gouging, grinding abrasion of parts hard to lubricate, can be eliminated entirely if sufficient load is applied to stop relative motion or if the parts oscillate enough to replenish lubricant on the surfaces; a critical, in-between amplitude of oscillation was found to be the most damaging.

Solid lubricants must be used in some bearings operating at temperatures at which petroleum or organic fluids decompose. Godfrey and Bisson[9], in studying the effect of powdered molybdenum sulfide on friction and wear, found that this solid lubricant was most effective when bonded to the bearing surface by a baked film of varnish or resin. A constant supply of molybdenum sulfide was thus kept on the contacting surfaces. This same National Advisory Committee for Aeronautics group under Bisson[11,21] has shown that synthetic lubricating fluids (chiefly esters) are more effective boundary lubricants (partial metal-to-metal contact) at high sliding velocities and higher bulk temperatures than petroleum oils of comparable viscosity. Lubrication failure at high temperatures appears to be due to inability of the fluid to maintain metal soap or iron oxide films on the bearing surfaces or to thermal instability of the chemi-sorbed metal soap film.

Indium has found favor as a component of certain automotive and aircraft bearings. Studies by Moore and Tabor[18] showed that the adhesion of indium to metals and nonmetals is closely related to the coefficient of friction. They also found that the frictional mechanism for sliding nonmetals is related to the cold-welding process characteristic of the friction of metals. Rabinowicz and Rightmire[25] have made a fundamental study of the high friction and scuffing experienced in cutting operations on titanium. Because of the non-adherence of protective chemical films to titanium, this metal adheres to steel tools, and machining operations on titanium are difficult. Molybdenum-resin films described by Godfrey and Bisson[9] were found by Rabinowicz and Rightmire[25] to be successful lubricants for titanium.

The frictional properties of synthetic high polymers have become of increasing interest because of their widespread use as fibers and protective coatings. Frictional forces in sliding plastics were found by Shooter[27] and Shooter and Tabor[28] to be similar to those in metals, and shearing was found to occur within the bulk of the plastic rather than at the interface. The exception was polytetrafluorethylene which gave a uniquely low coefficient of friction. This behavior probably results from the adhesion of TFE particles to the opposing surface followed by shearing at the interface because of the weak forces between —CF_2-groups.

Bowers, Clinton, and Zisman[2], in an investigation of the frictional properties of a series of unplasticized derivatives of polyethylene containing various proportions of fluorine and chlorine, found that substitution of chlorine for hydrogen in polythylene caused an increase in friction, while substitution of fluorine caused a decrease. The effect of chlorine was found to predominate when both fluorine and chlorine were substituted. Extremely thin films of polytetrafluorethylene on metal backings gave very low friction values, a property which is applicable to the dry lubrication of bearings in aircraft instruments, guided missiles, submarines, and torpedoes.

The life of cutting tools in machining operations influences manufacturing costs to a considerable degree. Present methods of evaluating cutting lubricants and operational factors on tool life, either by microscopic examination or by hours to tool failure, are slow and inaccurate. Merchant and coworkers[16,17] have applied tracer methods to this problem with the result that accuracy is increased three to five times and test duration cut to minutes, or even a few seconds. Special tool bits are irradiated in the Oak Ridge pile and then used in specially built test equipment to evaluate tool life. It was found that over 90% of the radioactive particles from the tool clung to the chips, and by counting gamma (or beta) radiation of the weighed chips, tool wear rate was measured in a matter of minutes. Radio-activity measurements were found to parallel results obtained by standard procedures, but with greatly increased accuracy.

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Section News

BALTIMORE, November '54. John Boyd (ASLE President), Westinghouse Electric Corp., presented a paper entitled "Sliding Type Bearings," illustrated with slides and a color film, demonstrating the action of journal and pivoted pad thrust bearings.

January. L. B. Sargent, Jr., Aluminum Co. of America, presented a paper entitled "Friction Reduced Through Practice." (Submitted by J. E. Buchanan, Sec'y.)

CHICAGO, November '54 W. C. Bauer, Briggs Filtration Co., presented a paper entitled "The Maintenance in Service of Lubricating Oil Quality by Filtering, Centrifuging, Reclamation." (Submitted by A. B. Wilder, Sec'y-Treas.)

FORT WAYNE, November '54. J. E. Shideler, Jr., International Harvester Co., presented a paper entitled "Clarification of Oil Contaminated Waters," illustrated with a color film, and followed by a tour through the company's Fort Wayne Works waste treatment plant. (Submitted by J. W. Buckner, Program Chrmn.)

INDIANAPOLIS, November '54. Election of Officers (see ASLE Directory), followed by an address by H. E. Nicholson, The Texas Co., entitled "Fundamentals of Fluid Lubrication."

Based on a planned course of lubrication fundamentals, the **December** meeting covered the subject of "Lubricating Oil Types & Testing," presented by a representative of the Socony-Vacuum Oil Co., Inc.; the **January** meeting embraced the topic of "Greases, Types, Application & Testing," by a representative of the D-A Lubricant Co., Inc.; with succeeding meetings to cover the following: **February**—"Oil & Greases," by L. E. Danford of Western

(Continued on Page 58)

KANSAS CITY, October. Thirty-eight registrants attended the 3-day Condensed Course in Lubrication Engineering sponsored by the ASLE Kansas City Section. Pictured above, from left to right, are: (1st row) R. R. Matthews (Moderator), H. H. Brunner, C. A. Bailey, Dr. H. E. Mahncke, L. M. Reiff, T. N. Bath; (2nd row) W. R. Johnston, W. I Best, P. Cuckworth, D. Howell, E. E. Gebhardt, G. J. Clark, D. L. McCormick; (3rd row) C. Heith, C. Pope, R. Theiss, C. E. Gore, R. D. McCrum, M. M. Mitchelson, R. H. Cottrell, R. P. Schmitz, W. D. Herndon, W. C. Lindsey, J. Brizzolara; (4th row) F. W. Starkey, V. L. Thompson, C. T. Kleinsorge, R. E. Shepherd, J. M. Crocker, G. Williams, H. A. Dalby, R. Jolly, R. J. Seybried, R. A. Gibson, L. W. Beal, E. R. Allgeyer. (Submitted by L. M. Reiff, Chrmn.)

PITTSBURGH, November. Fifty persons attended the 2nd Lubrication Engineering Conference on the "Selection of Lubricants," sponsored jointly by the Mechanical Engineering Department of the University of Pittsburgh and the ASLE Pittsburgh Section. The Conference Committee included A. A. Raimondi (Chrmn.), C. A. Bailey, J. Boyd (ASLE President), Prof. N. L. Buck, J. L. Duchene, E. S. Francis, Dr. H. E. Mahncke, M. C. Miller, M. Parker, L. B. Sargent, and D. W. Sawyer. Papers were presented by C. R. Schmitt, E. F. Houghton & Co.; Dr. W. J. Wojtowicz, H. A. Montgomery Co.; T. Badger, Westinghouse Electric Corp.; S. D. Craine, W. A. Jones Foundry & Machine Co.; M. Ehrlich, American Lubricants, Inc.; Dr. M. E. Merchant, Cincinnati Milling Machine Co.; and A. E. Roach, General Motors Research Labs. (Submitted by A. A. Raimondi)

A SIMPLE DEMONSTRATION OF FLOW TYPE IN GREASES

by H. E. Mahncke* & W. Tabor**

The laminar flow properties of oils and greases when passing through tubes of circular cross section were studied by a method in which the radial velocity distribution could be directly observed. The velocity distribution, deduced from theory for two kinds of materials, was compared with that found experimentally. The expected distribution for a Newtonian fluid was found for oil, while the grease exhibited the velocity distribution characteristic of a Bingham Body. So-Called "Plug-flow" is observed in the latter case.

A great deal of work has been done in the past few years on the problem of the rheological properties of grease [1a, 1b]. It was recognized quite early that greases did not obey the simple Newtonian relation:

$$\text{Shearing stress } (s) = \mu R \qquad (1)$$

in which (μ), the viscosity, is independent of the shear rate (R). Accordingly, most of the work on this problem has been concerned with establishing empirical correlations between an apparent viscosity and the rate of shear. Wilson and Smith[1a] have attempted to include in such a correlation a term to take account of the dependence of viscosity on the time of stress application (thixotropy).

Grease rheology would be on a sounder basis if it could be shown that some general rheological relationship based on definite physical concepts was applicable to grease type materials. An expression of this sort which is immediately suggested is the Bingham equation:[3]

$$s = \mu_{pl} R + s_y \qquad (2)$$

where μ_{pl} = *plastic viscosity*
s_y = *yield stress*

This equation contains two constants for the characterization of a given material but, barring the Newtonian equation (one constant), it is the simplest one available and therefore deserves consideration. It would be quite advantageous if a grease could be characterized by these two constants having quite specific physical meaning instead of by the specification of empirical numbers having no fundamental significance. For example, Milne[2] has recently presented an analysis of a slider bearing assuming a Bingham Body for the lubricant. Such a development has the possibility of improving considerably the rationale of grease lubricated bearing design and operation.

Singleterry and Stone[4] have studied the applicability of the Bingham relation to two samples of grease and found good agreement. This work was brought to the authors' attention while the present study was in progress and is, so far as known, the first experimental evidence that laminar flow of greases would be of the Bingham type. These authors measured the rate of flow of grease through a capillary type instrument and analyzed the data by means of the Buckingham equation for flow rate of a Bingham Body.[3]

$$Q = \pi r^4 \Delta p / 8 l \mu_{pl} [1 - \tfrac{4}{3}(2ls_y/r\Delta p) + \tfrac{1}{3}(2ls_y/r\Delta p)^4] \qquad (3)$$

where Q = volume rate of flow through a tube of length (l) and radius (r) under a pressure head (Δp). The usefulness of this method is somewhat impaired by the complexity of the calculations involved in analyzing the data, although some short cuts have been worked out by the authors.

The present approach to the problem has been to study the radial velocity distribution of a grease flowing through a tube with a circular cross section. The distribution so determined can then be compared with that predicted for a Bingham fluid, which is[3]

$$v = 1/\mu_{pl}[r^4 - r_z^4)\Delta p/4l - s_y(r - r_z)] \qquad (4)$$

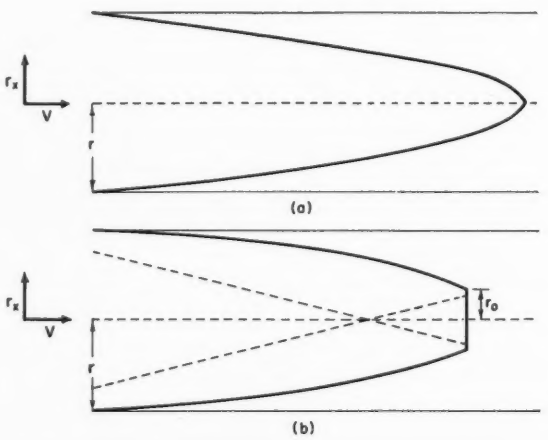

Fig. 1a (top). Velocity distribution for a Newtonian fluid.

Fig. 1b (bottom). Velocity distribution for a Bingham body.

*Westinghouse Research Laboratories, East Pittsburgh, Pa.

**Harvard University, Dept. of Chemistry, Cambridge, Mass.

This paper was contributed by ASLE, and presented at the 1st Annual ASLE-ASME Lubrication Conference, Baltimore, Maryland, October 18-20, 1954.

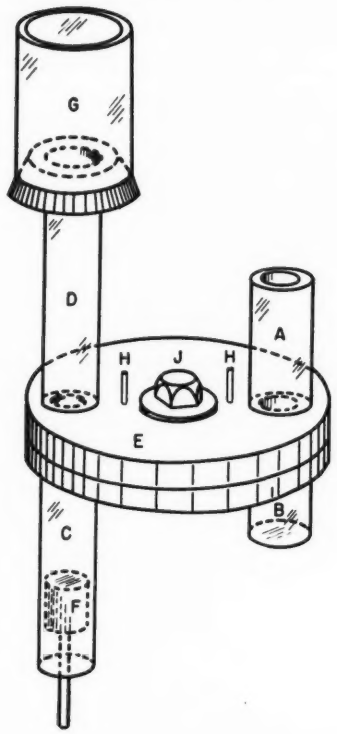

Fig. 2. Apparatus used in the oil flow experiments.

where v = the velocity at any radius r_x. The form of the velocity profile for a Bingham fluid is shown in solid line in Fig. 1 compared with that of a Newtonian fluid. In the Newtonian fluid the velocity is related to the radius by the familiar expression

$$v = \triangle p(r^2 - r_x^2)/4l\mu \qquad (5)$$

The truncated portion of the parabola is the diameter of a cylinder of material moving through the tube in which no shearing occurs. This is one form of the well known phenomenon of "plug flow."

A study of velocity distribution offers the possibility of more direct comparison of materials, lesser complexity in data analysis, and of more detailed investigation of the relation of structure to rheological properties. The present work constitutes an initial step in this direction.

EXPERIMENTAL. The method used for the determination of the velocity profiles was based on an apparatus designed by MacInnes and Brighton[5] for the study of transference numbers in solutions of electrolytes. The essential feature is the creation, in a fluid, of a definite boundary surface whose distortion can be observed as a stress is applied to the fluid.

Referring to Fig. 2, the apparatus consisted of two brass discs (E) ½" thick by 6" in diameter in which were fastened 1" I.D. tubes, (A,B,C,D). The tubes extended through the plates and met edge to edge, since the opposing surfaces of the plates were ground flat after the tubes were inserted. The alignment of the tubes was maintained by the dowel pins (H) while the bolt (J) holding the plates together provided an axis of rotation for the discs. For the purpose of checking the apparatus, it was decided to first make some tests with a material known to be a Newtonian Fluid; e.g., a lubricating oil, and so glass tubes were used.

To charge the apparatus, tube (A) is placed over tube (C) where perfect alignment is not required. Tubes (A) and (C) are then filled with a colored oil and tubes (B) and (D) with an uncolored oil of the same viscosity. The upper plate is next rotated 180° with respect to the lower one, bringing tubes (C) and (D) into alignment and the dowel pins put into place. A teflon piston (F) is then raised in (C) by means of a power driven screw, which forces the dyed oil to flow into the clear oil making the velocity distribution for axial flow visible. This is by virtue of the fact that, in rotating the disc through 180°, a plane or sheared boundary between the colored and uncolored oil is formed normal to the direction of motion. Barring diffusion across the boundary, subsequent axial motion will cause it to assume the shape corresponding to the distance traveled by each point in the boundary, and hence corresponding to the velocity experienced by each point. The container (G) on top of the tube (D) catches the

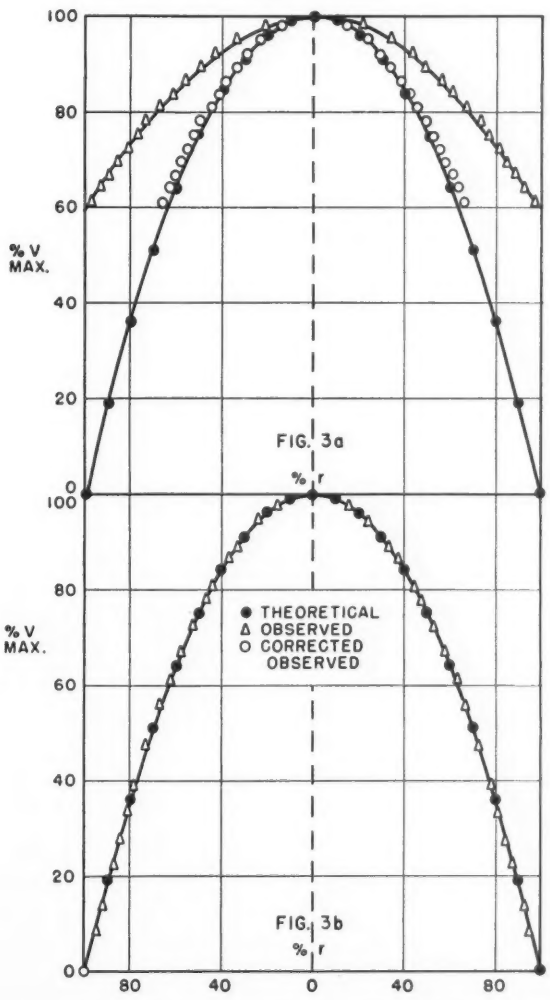

Fig. 3a (top portion); Fig. 3b (bottom portion).

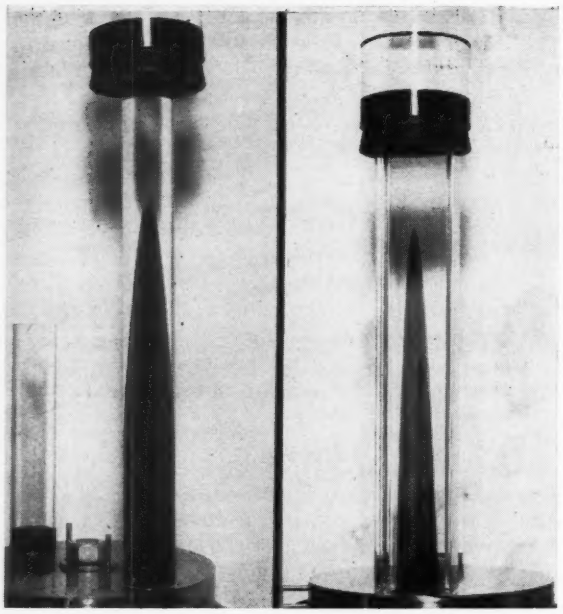

Fig. 4a (left). Parabolic profile showing lens effect.

Fig. 4b (right). Optically corrected profile.

overflow. The long tubes (C) and (D) are both 12″ while the short tubes are 6″.

Since greases are opaque and obscure the boundary, the flow process cannot be observed in glass tubes. The technique is, therefore, modified by providing 1″ diameter brass tubes, which are split lengthwise, to replace the glass tubes. They are threaded on one end and can be screwed into the brass plates with their ends flush with the opposing surfaces. The two halves are held together by means of close fitting brass rings slipped over the assembled tube. One tube is filled with a dyed grease while the other is filled with a normal one and the sheared boundary is formed and moved in the manner just described. The tube that holds the distorted boundary is then unscrewed from the plate, cooled either in dry ice or a refrigerator to stiffen the grease, and finally split in half using a tight wire for a cutter. Each half of the tube then exhibits a profile of the flow pattern.

For quantitative measurements, photographs were taken of profiles made either in the glass or the split tubes. The negatives were measured with a traveling microscope and the profiles reproduced on a plot of radius vs. constant x velocity.

RESULTS & DISCUSSION. The first experiments were made with oil in the glass tubes to see if, with a known Newtonian Fluid, the parabola predicted by equation (3) could be realized. A water-white paraffin oil of about 60 centistokes viscosity at 100 F. was used for this purpose. 0.05% of a red gasoline dye was added to give the colored oil, which caused a negligible change in the viscosity.

When measurements on the boundary were first made, considerable distortion of the image was encountered due to the lens effect of the oil filled tube. After applying the necessary correction the displaced boundary profile gave an excellent fit to a parabola of the type predicted by equation (5), although the measurements could not be extended to more than 70% of the tube radius. No dissymmetry was found in the shape of the parabola, although some departure from parabolic behavior is noted as the magnitude of the correction increases. Fig. 3(a) shows typical data obtained in this way. Since the measurements on the negatives are made in arbitrary units, the data is reproduced as % of the tube radius and % of the maximum velocity as indicated by the maximum displacement of the parabola.

Since it was found that the index of refraction of the oil was the same as that of the glass, the necessity for making the above mentioned correction was obviated by putting a square tube, made of Lucite plates cemented together, around the glass tubing and filling the space with oil. This eliminated the lens effect and allowed a true picture to be made. Measurements could be made on these pictures up to 100%R and gave an even better fit to a Newtonian parabola than the previous data. Typical results are shown in Fig. 3(b). These lead to the conclusion that the conditions for Newtonian flow are completely realized in this experiment. Fig. 4 shows the distorted and undistorted profiles from which measurements were made.

Incidentally, the apparatus is useful in other stud-

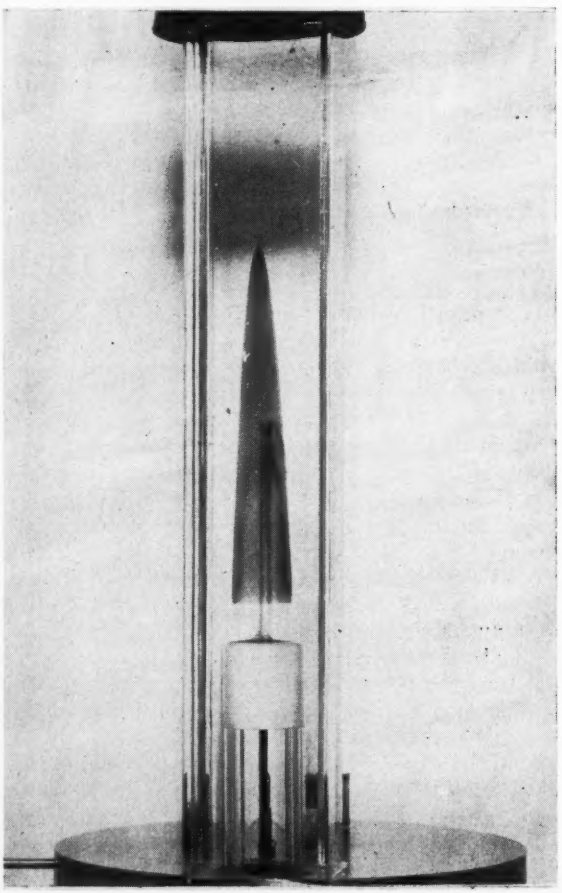

Fig. 5. Oil flow pattern with original boundary near driving plug.

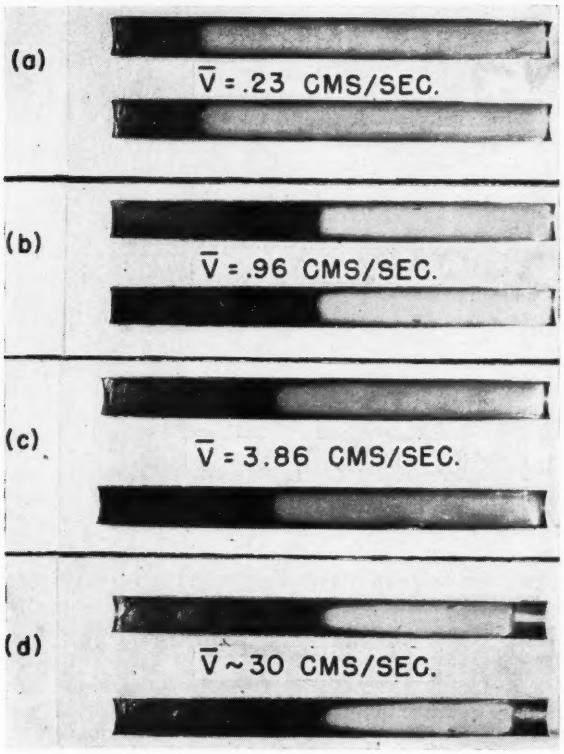

Fig. 6. Grease flow patterns (Grease A).

ies of flow problems. For example, in the derivation of the equations for Newtonian flow, the ideal case of a fluid flowing through a pipe with no end effects is assumed, so that there is no interaction with the agent producing the flow. The present apparatus afforded an opportunity of studying the flow in the vicinity of the driving plug to see how its proximity affected the pattern. In one experiment, a sheared boundary was formed with the plug, ½" below it. Upon moving the plug, it was observed that a parabolic front obtained immediately. After the parabola was well developed, however, (about 5 inches motion of the plug) some unusual features were observed as can be noted in the photograph (Fig. 5). White oil had moved radially inward and then upward to form another parabola within the colored one. The colored oil had moved away from the plug with the exception of a very thin layer immediately adjacent to it and a thin column at the center rising vertically and joining the main body of colored oil. There is, thus, grounds for belief that radial laminar flow occurs quite close to the plug, but with a continually changing direction until true axial flow is established, a short distance from the plug. Since the plug was always 10 inches or more from the sheared boundary in most of the experiments, there is little reason to suspect that the plug had any effect on the shape of the velocity profile.

Having established the validity of the experimental method, flow patterns of lubricating greases were next made. Four different greases were used in these experiments as follows:

Grease A—A mineral oil vehicle of 32 centistokes viscosity at 100 F., thickened with 12% of a sodium soap and 3% of a calcium soap.

Grease B—A diester oil thickened with a lithium soap.

Grease C—A silicone oil thickened with a lithium soap.

Grease D—Silicone stopcock grease. A silicone oil thickened with amorphous silica.

In all of the experiments with grease, time of motion and displacement of the driving plug were recorded so that absolute velocities could be established as well as rate of volume flow. During the cooling of the grease prior to cutting the column, some lengthwise contraction occurred, which was corrected for in the final measurements. Photographs of each grease were made for a series of constant plunger velocities ($\overline{V}$) and are shown in Figs. 6, 7, 8 and 9. In all cases, the phenomenon of plug flow is clearly apparent in that the plug radius decreases with increase in $\overline{V}$. As can be shown from the elementary theory of the Bingham Body, this is to be expected. From equation (4) maximum velocity is found at

$$dv/dr = 0, \text{ whence plug radius } r_o = 2ls_y/\Delta p \qquad (6)$$

and since

$$\overline{V} = Q/\pi r^2$$

then from Eq. (3) Δp increases as $\overline{V}$, and hence (6) r_o decreases. Fig. 10 shows, also as expected, that the plug radius does not change with distance as long as $\overline{V}$ remains constant.

In the treatment of the data obtained in these experiments, a somewhat different procedure was followed than that used for the oils, since with a Bingham Body expression it is not possible to eliminate the constants by using fractional values of the radius and velocity. True velocities were obtained as the quotient of the

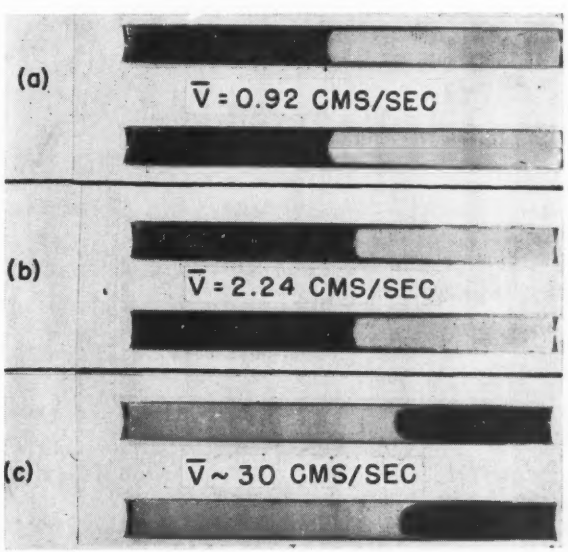

Fig. 7. Grease flow patterns (Grease B).

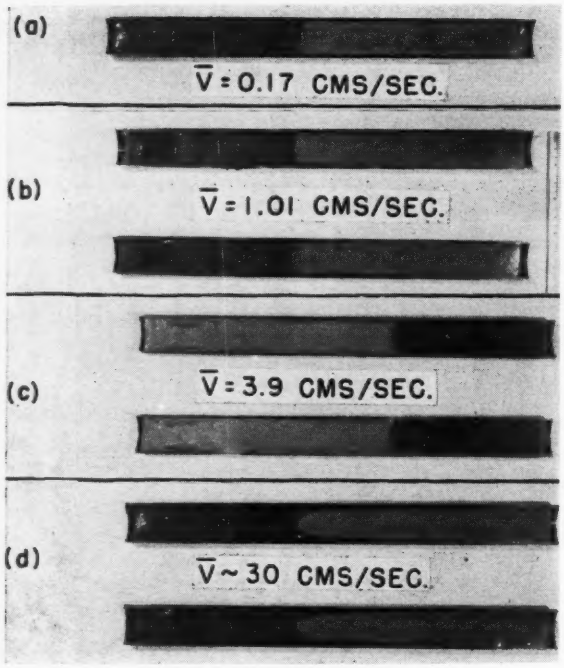

Fig. 8. Grease flow patterns (Grease C).

measured distance and time. Since the tube axis could not be conveniently located, the radial distance of the boundary from the tube wall was measured. ($X = r\text{-}r_x$) Fig. 11 shows data obtained in this way for the grease shown in Fig. 7(c). (Mixed base grease, plunger velocity 3.86 cms./sec.).

The calculated values of Fig. 11 were obtained as follows. Since, for a given experiment, all the parameters except v and r_x are fixed we may use equation 4 in the form,

$$v = k_1(r^2 - r_x^2) - k_2(r - r_x) \tag{7}$$

and

$$\textit{plug radius } r_o = k_2/2k_1 \tag{8}$$

$$v_{max} = k_1(r - r_o)^2 \tag{9}$$

Using equation (7) to solve for k_1 and k_2 by substituting experimental values of v and r_x *is* straightforward but tedious arithmetically. A more convenient method and one which is more sensitive is as follows:

Let $X = r\text{-}r_x$, which is the quantity measured radially. Substitution of this in (7) then gives

$$v/X = k_1(2r - X) - k_2 \tag{10}$$

When v/X is plotted against $(2r\text{-}X)$ a straight line plot should be obtained if the material used fits the Bingham Body expression. Also, when $v/X = o$, we have for the intercept

$$(2r - X)_o = k_2/k_1 = 2r_o \tag{11}$$

This procedure has been carried out on the data for Fig. 11 with the results shown in Fig. 12. The one disadvantage seems to be that the extrapolation for the intercept is a fairly long one, though it seems valid on the basis of the straight line obtained.

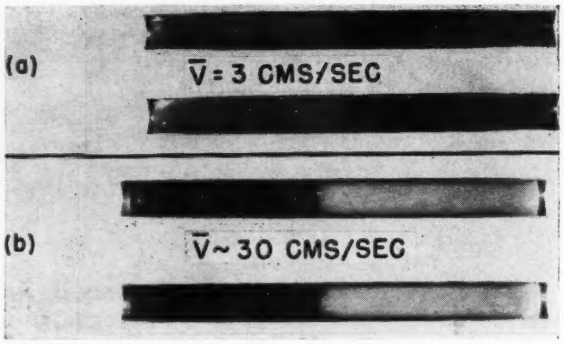

Fig. 9. Grease flow patterns (Grease D).

Using the constants obtained in this manner, points were calculated and plotted on Fig. 11 (open circles). The agreement is seen to be quite good except in the case of the location of the point v_{max}, r_o. This point is quite sensitive to the values of k_1 and k_2 chosen, so that accurate location will require more care in the experimental technique than was applied in the present series of tests. This, however, will not be difficult to achieve.

The interest in finding this point more exactly derives from the possibility of using this type of experiment to measure still another property of grease. As has been noted, the advancing front of the plug should have a plane face, but experimentally the front is observed to be convex. Let us denote by l_y the distance on the tube axis between the theoretical plug front (v_{max}) and the actually observed front. Then it can be shown[3] that

$$l_y = r_o\gamma/2s_y \qquad \text{or} \qquad \gamma = 2s_y l_y/r_o \tag{12}$$

where γ is the shear modulus of the grease.

Measurements of the sort described above were carried out on the other types of greases. However either the plugs were too wide, making X difficult to measure accurately, or r was not well enough known. Thus, while it was found that the Bingham Body relationship was approximately satisfied in each case, it was not possible to say how closely, so no further analysis will be attempted here.

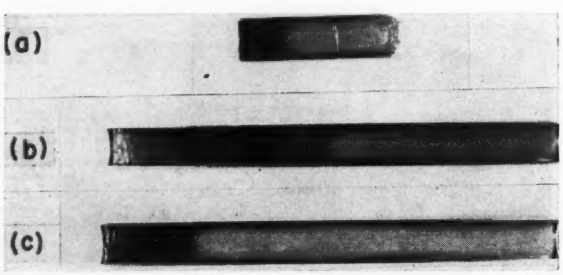

Fig. 10. Grease flow patterns const. velocity $\bar{V} = 0.3$ CMS.

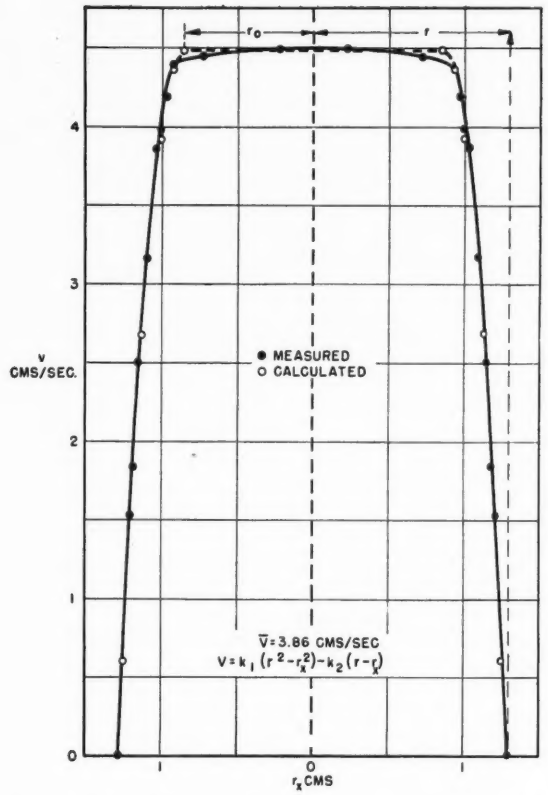

Fig. 11. Flow pattern for Grease A.

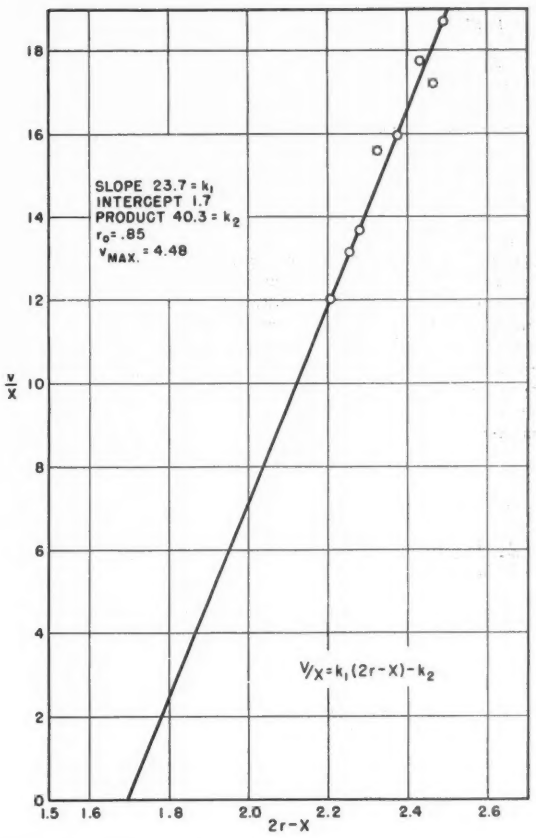

Fig. 12. Plot of function.

The significant feature of these results is that it is possible to make a direct study of the type of flow law or consistency law which a grease obeys. If, as appears now, most greases will obey the Bingham Body relationship, then it will be possible to characterize the material by the specification of two constants, i.e., the plastic viscosity (μ_{pl}) and the yield value (s_y). In some cases, also, the shear modulus (γ) may be of interest.

It is clear, however, that much more work will have to be carried out, either by rate of flow methods as exemplified by the study of Singleterry and Stone, or by modifications of the present technique in which control of the experimental parameters is effected, before any overall generalizations are attempted. In this connection, two points of considerable interest may be mentioned. The first is the effect pointed out by Wilson and Smith, and others, of thixotropy. In this case there is the possibility of including a time dependent factor in the Bingham relation. The second point is the possibility that some other functional relationship between rate of shear and shearing stress may apply to greases. Studies of the sort discussed here will be quite helpful in further exploration of these problems.

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COMMENTARY by C. R. Singleterry, Naval Research Laboratory. The authors are to be commended for their contribution to our understanding of grease rheology. The experimental procedure is simple but ingenious and deserves to be extended to greases for which the two characterizing constants can be expected to vary as widely as possible.

The data presented give further support to the Bingham Body model for lubricating greases. The advantage of a two-constant characterization of the rheological properties of a lubricating grease, as compared with the present empirical makeshifts, are substantial whether one is concerned with the functioning of greases or merely with their specification for purchase. Even if greases should turn out to conform only approximately to the model, the practical characterization of their flow properties by two constants having clear-cut physical significance would be an important advance. It may be hoped that this work brings us closer to the adoption of a rational method of reporting grease consistencies.

COMMENTARY by N. Marusov, Gulf Research & Development Co. The major contributions by the authors are the development of an inexpensive apparatus and a demonstration of the ease with which the velocity distribution of Newtonian as well as non-Newtonian materials may be obtained experimentally. The apparatus and the techniques described are not necessarily limited to the study of lubri-

cating greases or Bingham type materials as the title may imply.

We were particularly interested to note that the advancing front of the plug was convex instead of having a plane face as it would have if the material were strictly Bingham in character. Our own experiments with a capillary-type viscometer have indicated that lubricating greases are not strictly Bingham, but may fall in the classification of a general plastic which would give a convex face.

The authors also suggest several possible fields of exploration which, we agree, would be of real value and interest. We would add one more to the list; namely, the development of graphical methods for predicting flow-pressure behavior of non-Newtonian materials from the velocity distribution of these materials. A planimeter would give the necessary integration regardless of the functional relationship of shear stress and rate of shear. The authors are to be commended for presenting a technique which gives the velocity distribution in a direct and easily performed manner.

AUTHORS' CLOSURE. Mr. Marusov's comment relative to graphical methods for treatment of flow data of non-Newtonian materials has been given some thought by the authors. Indeed, the original purpose of the experiment was to study the shear stress, rate of shear relation by this means and to see whether or not the velocity distribution could be subjected to graphical differentiation. However, the most significant result so far has been the agreement with the Bingham relation.

A point which was perhaps not established clearly in the paper is that the relation for velocity distribution, Eq. (4), does not completely define the velocity distribution curve, but only has physical significance for $r > r_x > r_o$; i.e., in the region in which shear occurs. The complete curve is obtained from consideration of symmetry and the fact that no shear occurs for $r_x < r_o$. The dotted portions of curve 1 (b) are the portion of the analytical expression which have no significance. Our thanks are due to Professor Blok for emphasizing this point.

New Products

GUM TEST APPARATUS. A new gum test apparatus for ASTM D-381 tests is now available designed to run five gum tests simultaneously in motor, aviation, and jet fuels with air jets or superheated steam jets. Unit is self-contained, consisting of aluminum bronze bath, built-in steam superheater, adjustable thermo-regulator, and insulated stainless steel jacket; flowmeter and removable jet nozzles are also included. (Bulletin 1220, Labline, Inc., 217 N. DesPlaines St., Chicago 6, Ill.)

ADDITION AGENT. An addition agent, Santolene J, which enhances the color stability of fuel oils, inhibits rusting, and reduces organic sediment formation, is now available in commercial quantities. A concentration of 25 pounds of this new product per 1000 barrels of fuel oil will stabilize the color and reduce sedimentation of most base fuels. Example: After 12 weeks storage at 110 F., oil treated with this dosage had an NPA color of D-5; sediment measured 27 milligrams per liter. Untreated oil's color had darkened to D-6, and sediment per liter amounted to 103 mg. (Monsanto Chemical Co., St. Louis 4, Mo.)

FERULOK FINISH. Steel Ferulok tube fittings are now being produced with a permanent, attractive black coating which acts as built-in lubrication. (The Ferulok is a flareless type fitting which depends on the nut forcing a ferrule to bite into the tube to provide a pressure-tight connection.) The built-in lubrication of the new finish reduces considerably the wrench torque required to make-up the fitting, and provides easy reassembly of the joint without the use of oil. (Parker Appliance Co., Tube & Hose Fitting Div., Cleveland 12, Ohio.)

ULTRASONIC EMULSIFYER. Model EM 750, third in a line of ultrasonic emulsifying machines, has been designed to produce a pure emulsion of oelic acid in water at the rate of 4 gal/hr., using no stabilizer. Emulsifying action results from the intense cavitation forces of focused ultrasonic waves which produce a state of fine division of one liquid into another. The division is in the lower-micron and sub-micron range which helps to insure permanency and dilutability. (McKenna Laboratories, 2503 Main St., Santa Monica, Calif.)

EP SPRAY LUBRICANT. Spray-Lube, a new extreme pressure lubricant designed to spray a superior, heavy duty lubricant on open gears, open chains, valve stems, etc., without pre-heating and then applying with brush or paddle, has been developed to encourage proper lubrication of machinery that might otherwise be neglected. It can be used in temperatures too high for normal grease or oil, will not drip, or throw off running gears, and makes possible lubrication of parts a brush or paddle cannot reach. (Pressure Products Co., P. O. Box 342, West Chester, Pa.)

AUTOMATIC DRAIN. Intended for installation on receiver tanks, air line sumps, drain legs, pneumatic storage tanks, etc., the new Wilkerson sump drain (Model 150SD-4) automatically collects and discharges all moisture, free oil, and contaminants at any line pressure from 10 to 250 psi. Measuring 5" dia. by 8½" high, and made of cast aluminum, the unit is threaded for ½" pipe nipple on the inlet and can automatically collect and discharge over 400 gallons of water per hour. (Wilkerson Corp., 3401 S. Platte River Dr., Englewood, Colo.)

CONSTANT LEVEL OILERS. Designed to feed a complete range of oil from very light to 600W, new large capacity constant level oilers automatically maintain a constant level of oil in large ring or ball bearing chambers, pump and gear housings, textile comb boxes, etc. Large surge capacity is provided for any back surge caused by large amount of oil draining off mechanism when machine is stopped. Available in 10 oz., 1 pt., 1 qt., and 2½ qt. Lucite plastic or Py-

(Continued on Page 57)

FOAMING
OF A HEAVY-DUTY LUBRICATING OIL

by W. M. Pugh* & L. M. Tichvinsky**

A series of foaming tests were conducted by the bubble method and some of the factors controlling the reproducibility of the test were examined for the case of a heavy-duty lubricating oil. The method is sensitive to temperature variation during the course of the test, to variation of the rate of air flow, and to the previous history of the oil sample. In addition, it was found that the porous stones, even when new, have varying degrees of permeability. Recommendations for improving the reproducibility of the bubble test for foaming are given.

Introduction. Foaming characteristics of lubricating oils have become important in the last few years, particularly with respect to the compounded oils widely used in Diesel engines, the lubrication of aircraft engines at rarefied atmospheres and oils for use in automatic hydraulic transmissions in automobiles. Obviously, oil foaming during operation in these instances can not be tolerated. Bearing failures will occur in the two former cases and in the latter, the hydraulic transmission will become inoperative with any appreciable amount of air-oil foam[1].

An example of excessive foaming is shown on the photograph of Fig. 1 when a compounded SAE 50 lubricating oil was tested during a stability test conducted in accordance with Federal Specifications, Method 345.1.1[2].

A.S.T.M. and Federal Specifications for foaming tests were first described at the end of World War II and recommended for the evaluation of foaming tendency and foam stability of crankcase oils at specified temperatures[2].

Reasonable reproducibility of tests conducted by various agencies is an important feature which enables one to correlate and extrapolate data previously reported in technical literature. The apparent lack of reproducibility of foaming tests prompted us to make a series of experiments to clarify a few points connected with standard foaming tests[2]. Consequently, the present note describes a few salient points pertaining to the theories of foam formation and foam decay, together with results of several experiments made with a heavy-duty oil. On the basis of the described experiments, recommendations are given for improvements of a foam test method.

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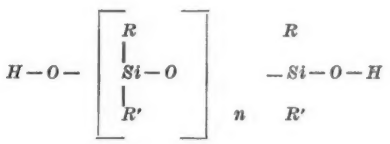
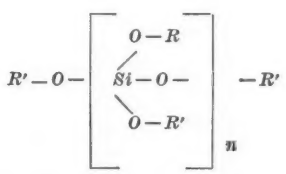
This paper was presented at the ASLE 9th Annual Meeting, Cincinnati, April 7, 1954.

General. Very active research was undertaken during the war for the purpose of elucidation of lubricating oil foam formation and of determining practical and efficient methods of foam suppression. Such research was conducted by many of the major oil corporations, various universities and independent laboratories, as well as by the U. S. Army and Navy. It appears that the Gulf Research and Development Company, Pittsburgh, Pennsylvania, and the Shell Development Company, Emeryville, California, independently filed patent disclosures almost simultaneously for lubricating oil foam suppressing processes incorporating the use of silicones. These, so far, proved to be the most active chemicals used as additives for effective foam control.

The liquid organo-siloxane and organo-silicate condensation products were found to be particularly effective. The representative molecular structure of such chemicals is as follows:



One of many possible organo-siloxane condensation products.



Probable structure of an organo-silicate condensation product.

Where n may be unity or more, R — organic radical (alkyl, aryl, aralkyl, alkaryl or heterocyclic group), R' — similar or dissimilar radical[3]. As hydrocarbon lubricating-oil foam-suppressing agents, the most effective compounds are those possessing low interfacial tension toward the oil, low surface tension, low solubility in oil, and those which are stable and finely dispersed in oil.

Suitable silicones are so effective in suppressing foam that such small proportions as from 10 to 100 parts per million by weight may be sufficient

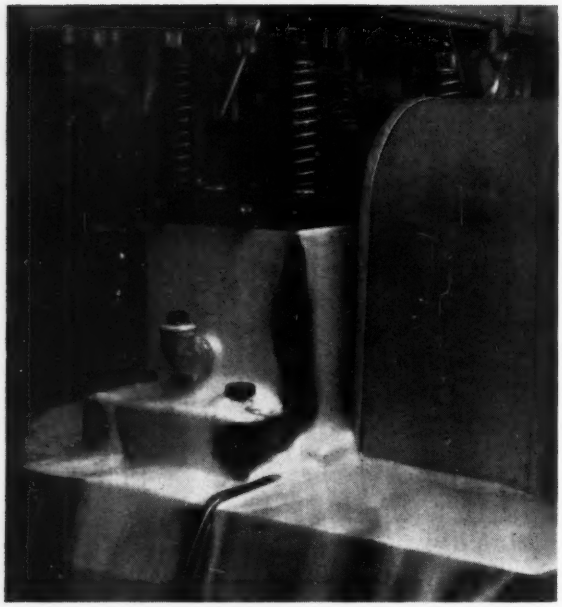


Fig. 1 Foaming of a lubricating oil in the navy work factor machine at 3000 rpm, 150 psi and 250 F.

for long periods without adversely affecting other important physical properties of a lubricating oil[4].

Methods of Controlling Foam Formation. There are several methods for the control of lubricating oil foaming. The chemical method appears to be effective, economical and flexible. Other methods, especially mechanical ones, which may be numerous, usually require changes in the design and, consequently, they are in most cases impracticable.

A standard method for predicting the defoaming characteristic of an anti-foam agent consists in measurements of surface tension and selecting such agents which fulfill the following two conditions[5]:

$$(1)\ \sigma f + \sigma df > \sigma d \qquad \text{entering condition}$$

$$(2)\ \sigma f > \sigma df + \sigma d \qquad \text{spreading condition}$$

where σf — surface tension of foaming oil, σd — surface tension of anti-foam agent, σdf — interfacial tension between anti-foam droplet and foaming oil.

Theory of Foam Formation and Foam Inhibition. It is beyond the scope of the present paper to go into detailed analysis of different existing theories of foam formation and foam inhibition. Therefore, only a few remarks on these subjects are presented here. Although there are several theories explaining the mechanism of foam formation, it is believed that a combination of several of them should serve to account for analytical elucidation of the foaming phenomenon.

Foam is considered as a heterogeneous system, consisting of a dispersed gas phase, separated by films of continuous liquid phase. For stable condition of such a system a continuous force must be maintained in the liquid phase and of such magnitude as to balance the force due to gravity drainage.

The physico-chemical phenomenon of oil foaming has been described in several publications[6,7,8,9]. From the review of these one may conclude that some of the following conditions are necessary for a given liquid-gas system to produce a stable foam: (a) positive or negative adsorption of some solute at the liquid-gas interfaces, (b) a definite surface structure consisting of oriented molecules, or a gel produced by hydration or other chemical action and forming a solid or plastic-like surface, (c) a surface viscosity substantially greater than the bulk viscosity, and, (d) a surface tension which is varying rapidly with changes in concentration.

There are also several theories explaining the mechanism of foam inhibition and it is believed that chemical inhibition of foaming may be accomplished by more than a single mechanism[10].

Heterogeneous and stable foam may be disrupted by inclusion in the films of small droplets of an emulsified substance causing rupture of the films. This may occur through the incorporation of silicone compounds possessing a considerably lower surface tension than mineral oils. Foam inhibition may also be achieved by such factors as volatility, producing unequal concentration at the interfaces, contact of the foam with another liquid or vapor, and by mechanical or thermal shocks such as sudden and rapid expansion, contraction, or temperature change[6,10,11,12].

Physical properties of the lubricating oils are of importance when determining the foaming characteristics[13]. The relationship between foam stability and surface tension is rather complex[8,14,15,16,17]. A lowered surface tension would theoretically diminish the energy required to form a foam. However, foams may be produced from solutions in which the solute is negatively adsorbed at the surface, as well as from solutions in which the solute is positively adsorbed.

McBain and Robinson investigated the effect of adding various antifoaming compounds upon the surface tension of aviation lubricating oils. With the exception of silicone compounds, which lowered the surface tension from 31.65 to 22.72 dynes per centimeter, there was practically no change in the surface tension (it varied from 31.2 to 31.8 dynes per centimeter), although the foam stabilities of the modified oils were varied within wide limits[17]. For all practical purposes it was found that the surface tension decreased linearly as the temperature was increased from 20 degree C. to 110 degree C.[18].

Methods of Test and Test Results. Many attempts have been made to devise accurate foam-testing procedure and introduce units of foaminess [19,20,21]. All these were based on such postulates as: (a) the volume of liquid formed from the foam is a measure of bubble decay, (b) subsidence of foam follows a logarithmic law, and (c) the number of bubbles present at any given time is proportional to the weight of the foam. It became apparent that, due to the complexity of foam stability, regarded

as a specific property of a liquid, and due to the considerable time required to determine it by means of proposed testing procedures, it may be desirable to use some less exact but more readily determinable method. Such a method was devised and recommended by the Cooperative Research Committee of the Coordinating Research Council and is described as Method 321.1.1 in the Federal Specifications[2]. It is a bubbling method which appears to be best suited for experimentation with various grades of lubricating oils, keeping in mind, however, its inherent limitations which, in many instances, may explain apparent discrepancies and irreproducibilities of test results. In the described experiments the bubbling test such as briefly described in Federal Specifications[2] was selected for limited studies of foaming properties of a heavy-duty lubricating oil on account of the following factors: (a) The necessary apparatus may be constructed from standard laboratory equipment at low cost, (b) It provides an adequate head of foam with heavy lubricating oils, (c) Foams may be produced and measured at various temperatures and at varying air rates.

The test set-up is shown diagrammatically in Fig. 2. A 200 cc sample of the oil to be tested is placed in the one liter graduated cylinder, sealed at the top by a rubber cork, and placed in a transparent liquid bath. After bath temperature equilibrium has been obtained, air is forced to the cylinder via the drying tower, with regulation and control achieved by three oxygen-bottle type reducers placed in series, and via a U-tube manometer, which measures the pressure drop across a short section of glass capillary tubing used as an orifice. A standard porous stone sphere or cylinder (gas "diffusion stone" or gas "scrubber") is connected to the bottom of the air delivery tube and immersed in the oil sample in order to disperse finely the air supplied into small bubbles. The amount of air leaving the graduated cylinder is measured in a gas meter, which may be placed before or after the graduated cylinder.

Measurements are made of the initial oil level and periodically of the upper and of the lower foam interfaces after foaming has started, thus establishing a foaming rate. Zero time is assumed to be when the first bubbles of air appear on the surface of the oil sample. Foaming may be continued until the average height of the foam reaches a constant value, termed the "equilibrium foam height", and then the rate of foam collapse is measured in a similar manner after the air supply has been shut off.

A sample of SAE 50 base oil of 85 Viscosity Index and -5 F. Pour Point solvent refined at the Richmond refinery of the Standard Oil Company of California was provided by the California Research Corporation and used for the described foaming tests.

Considerable investigation has been undertaken in order to improve the reliability of bubbling methods of determining the foaming tendencies and numerous reasons have been put forth to explain possible inconsistencies and irreproducibilities of test results[13,23]. The condition of the porous stone and the rate of air flow have been found of importance in improving the reliability of air bubbling methods[24,25]. It has been shown that equilibrium foam heights are a function of both bubble size and air input rate. For a given rate, the smaller the bubbles, the greater will be the amount of foam produced. Considerable bubble coalescence near the surface of the stones was found at high rates of flow, thereby lessening the amount of foam produced, whereas at too low a rate of flow the air tends to channel through the large pores in the stone. In addition, considerable turbulence is produced in the foam column favoring additional bubble coalescence at high rates of flow. It is therefore recommended that the lowest air flow rate which produces a satisfactory head of foam without channel-

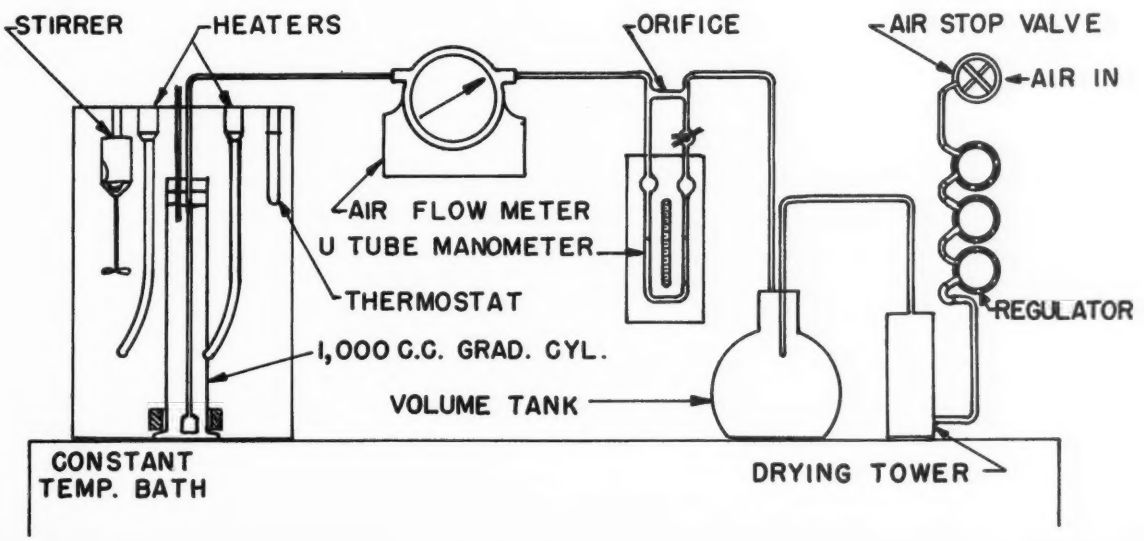


Fig. 2 Foam test set-up.

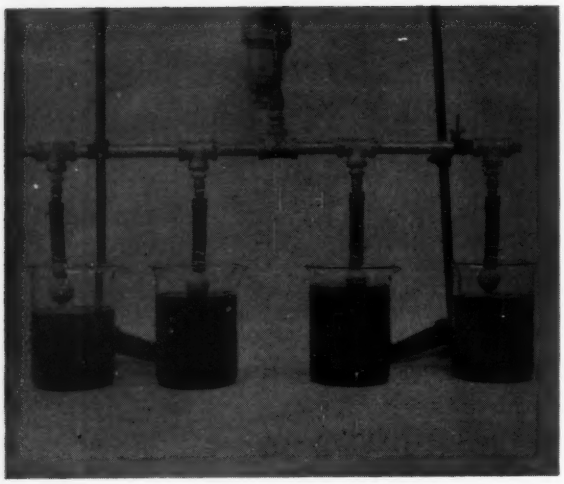


Fig. 3 Permeability tests of new porous plugs.

ing be used for measurements. Considerable variation was also found to exist between foam produced by different stones, other factors being constant.

A simple test was made for the purpose of verifying the uniformity of the permeability of porous stones. As shown on the photographs of Fig. 3, four porous plugs were connected to a header, filter and water line. Water was then forced through the plugs under approximately 90 psi pressure for a fixed time and collected in beakers placed under the plugs. The photograph shows the different volumes of water collected (dyed for clarity) when four new, unused spherical plugs were subjected to this test indicating various levels of water for all plugs.

Because of non-uniformity of porous plugs it may be advisable when conducting foaming bubble tests to consider testing a plug before and after a series of tests. This may indicate a possible contamination of the porous material by solids or failure to achieve the same degree of liquid saturation for both tests. The use of spherical or cylindrical plug should be specified because of different air rates.

Experiments were conducted to ascertain the cause of possible variation of the bubble test results, and the following additional variable appeared to be of significance: (a) Temperature variation during the course of the test, (b) Variation in rate of air flow, especially at the start of the test, (c) Length of time since previous aeration, if any, (d) Total amount of previous aeration of the sample.

It was found that the thermostat which controlled the bath temperature with a variation of ± 1 Deg. F. through the range of 85 to 200 Deg. F. was inadequate. Tests showed that a slow change in temperature of between one and two degrees could cause a variation in foam height of as much as 40 cc in a column of height 500 cc in the graduated cylinder, even though equilibrium foam values at the two temperatures might be indistinguishable. Consequently, all tests were conducted with manual temperature control, which resulted in a temperature variation of ±½ Deg. F.

An air rate of 0.1 cubic feet per minute (47 cc per minute) was found to be particularly suited to the equipment and to the 200 cc sample of oil used in all tests. Such a rate produced a satisfactory amount of foam throughout the temperature range of 75 to 200 F., with insignificant or no turbulence in the foam column. Slight differences in equilibrium height of the base oil for the same conditions of temperature and air flow rate which appear in the graphs are attributed to the different porous stone used in various tests. A single stone was, however, used throughout each series of tests.

Results of tests at various temperatures and same low air rate are presented on the graph of Fig. 4. Equilibrium foam volume is attained at approximately the same time of nearly 30 minutes. The rate of foam formation is higher for lower temperatures and diminished rather rapidly with increased temperature. As can be expected, more foam is formed at lower than at higher temperatures.

New oil and new oil which has been aerated will not perform equally in a foam tester. The effect of aeration is shown on the graph of Fig. 5, giving results conducted at 115 Deg. F. Curve A shows rate of foam formation for the first test with oil as received. If a similar test is conducted immediately after the first test, foam will be formed as indicated by curve B. This was a reproducible rate because consecutive tests yielded similar rates, within the accuracy of the equipment, regardless of the interval of time between tests, with the possible exception of very long periods of standing. This is indicated by curve C, which reveals a possible tendency of the oil to return to the original condi-

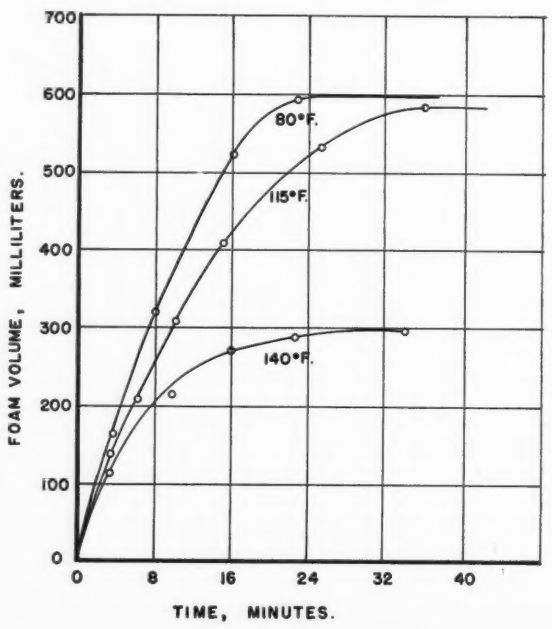


Fig. 4 Foam formations at various temperatures.

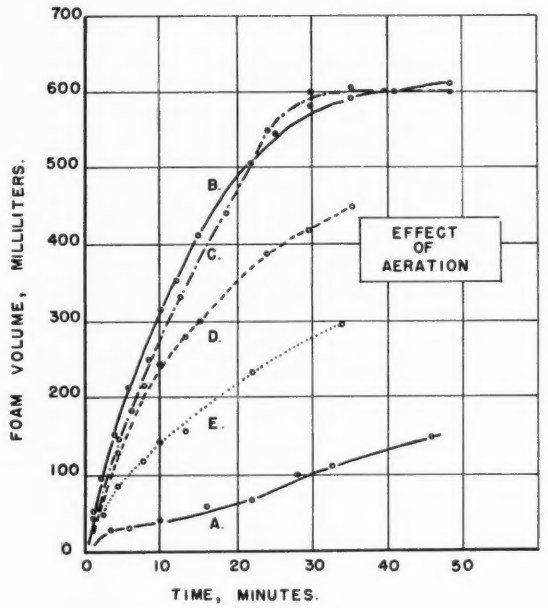


Fig. 5 Foam tests at 115 F. Key: A — Oil as received, test No. 1; B — Immediately repeated test; C — Test after two months; D — After evacuation at 1" Hg.; E — After chilling at 0° C.

tion after it has been allowed to stand idle for a period of two months. Curve D shows results of a test made after the aerated oil was subjected to vacuum of 1" mercury. A further decrease in the foam rate was obtained after chilling at 0 Deg. C. the aerated oil, curve E.

A more pronounced illustration of the effect of long standing is shown on the graph of Fig. 6, giving results conducted at room temperature. At this

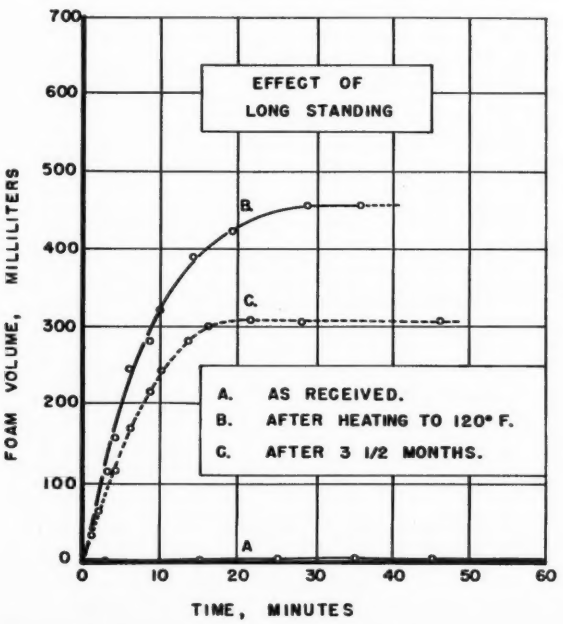


Fig. 6 Foam tests at room temperature 77 F.

temperature the oil did not foam at all, curve A. However, after this oil was heated to a temperature of 120 Deg. F. or higher, then cooled to room temperature, foaming tests resulted in a reproducible rate shown by curve B. After standing for a period of three and one-half months, the foaming tendency was reduced to that shown by curve C. Heating the sample to a temperature of 120 Deg. F. and cooling to room temperature restored the foaming rate to that shown by curve B.

In an attempt to detect the presence of emulsified air, measurements were made of specific gravity of the oil as it was received from the refinery and 45 minutes after aeration at room temperature. These results were not conclusive as no change in the value of the specific gravity could be detected with the use of the Westphal Balance. Very little data concerning the solubility of air in mineral oil appears in the literature. The effect of continued aeration at 200 Deg. F. on the foaming properties of the oil as determined by experiments is shown on the graphs on Fig. 7, giving results conducted at 80 and 114 Deg. F. As shown on these graphs, the average maximum foam volume attainable increased first with continued aeration. After a short period a condition was reached at which further aeration at 200 Deg. F. had no additional effect on the capacity of the oil to foam.

Since the completion of these tests, an extensive bibliography on foaming (nearly 1000 items) appeared in a book, "Foams—Theory and Industrial Application," by J. J. Bikerman[26].

Conclusions and Recommendations. These conclusions are drawn on basis of a few tests described which were performed on one sample of a com-

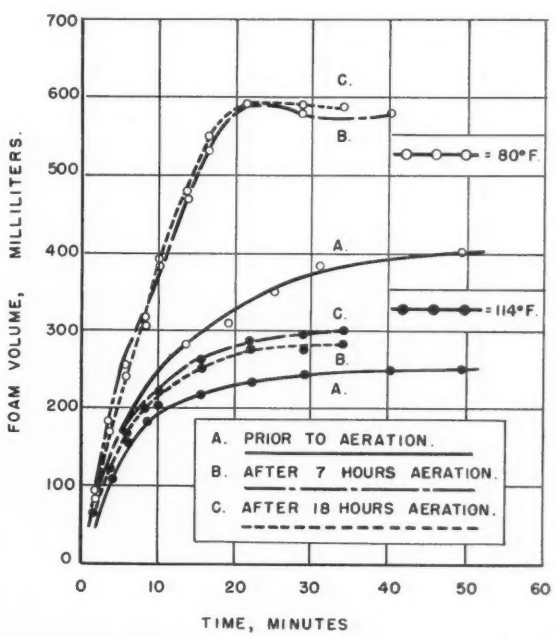


Fig. 7 Effect of continued aeration at 200 F., foam tests at 80 & 114 F.

pounded SAE 50 type lubricating oil only. However, it appears that similar conclusions would be proper for other type of compounded oils belonging to the heavy-duty category.

The past history of an oil sample with respect to both previous aeration and temperatures to which it has been exposed is an important factor to be considered when attempting to determine its foaming characteristics.

Since subjection of the oil to a vacuum results in reduction of foaming ability, it must be concluded that even small amounts, not detectable by specific gravity measurements, of dissolved or dispersed air promotes foam formation.

It is suggested that bubble foam testing will give more reliable results if preliminary aeration of all oil samples at 200 degrees F., were conducted prior to any foam test performed at lower temperatures.

The bubble method is particularly susceptible to air rate and temperature variations which should be controlled as accurately as practicable. Only carefully calibrated porous plugs should be used in order to ascertain their uniform permeability.

It appears from the tests performed and from the study of existing literature that of considerable practical importance would be a study of the compatibilities with respect to foaming of various commercial and approved Diesel engine oils. Foaming characteristics of silicone inhibited oils may vary and be adversely affected by such possible contaminants as fuel and its components, rust, salt water and grease, which may be present in minute amounts.

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Personals

At the '54 annual meeting of the National Lubricating Grease Institute **H. L. Hemmingway,** The Pure Oil Co., was elected President for the ensuing year, with **W. M. Murray,** Deep Rock Oil Corp., as Vice-President; Directors elected for a 3-year term include: **M. R. Bower,** Standard Oil Co. (Ohio); **A. J. Daniel,** Battenfeld Grease & Oil Corp.; **G. A. Olsen,** Sunland Refining Co.; **F. E. Rosenstiehl,** The Texas Co.; **W. H. Saunders, Jr.,** International Lubricating Corp.; and **B. G. Symon,** Shell Oil Co.

A. T. Olsson has been named an Assistant Sales Manager of Acheson Colloids Co., Port Huron, Mich.; **N. T. Bauer** has been appointed Service Engineer, with offices at 333 N. Michigan Ave., for the company's Chicago, Ill., sales territory.

Trabon Engineering Corp., manufacturers of lubrication systems, has announced the appointment of the **R. E. Kunz Co.,** 408 Broad St., Seattle, Wash., as its distributor covering the state of Washington and the panhandle of Idaho.

R. E. Dittoe, assistant secretary and a director of The Cleveland Worm & Gear Co., Cleveland, Ohio, has been appointed Sales Manager of the company's Worm Gear Div. in addition to his above duties.

C. A. Norgren Co., Englewood, Colo., has announced the advancement of **F. T. Goll** to Assistant Sales Manager.

(Continued on Page 56)

RADIANT HEATING OF LUBRICATING OIL

by R. H. Johnson*

Glass radiant heating panels have been found to serve well for maintaining oil reservoirs at slightly elevated temperatures without introducing localized hot spots in the oil. A working arrangement is described.

Before starting a large wind tunnel compressor after a prolonged shutdown, it was found necessary to provide lubricating oil to its bearings at a temperature of 110 F. Because room temperature at the installation site can be as low as 40 F., it was necessary to provide heaters to bring the oil from 40 F. to 110 F. and hold it at this temperature at all times when the bearings are not generating sufficient heat. However, to prevent damage to the oil, the heater surface temperature could not exceed 160 F. To maintain this relatively small temperature difference between heater and oil requires a very low-watt-density (watts per square inch) heater surface.

Most electric heaters used for heating lubricating oils are of the immersion type. These have surface watt densities so high that the resulting temperature of the heating surface can damage a sensitively-compounded oil. Without sacrificing heating rate, it is possible to decrease the surface watt density of the heater by increasing its surface area, but then the heater volume required for such an increase in surface area is so great that the immersion-type heaters prove impractical for this type of application.

The reservoir was 48 x 80 inches in area, 43½ inches deep, and had a 700-gallon capacity. It was found that by utilizing the bottom plate of the lube oil reservoir as a heating surface, enough energy could be passed into the oil to maintain its desired temperature and still keep the temperature differential between oil and bottom plate within limits. To heat the bottom plate of this reservoir, strip-heaters or some other type of radiant heater was required.

Strip-heaters could not be used because of their high concentration of heat in small areas, causing hot spots on the bottom plate directly above each heater. Although these hot spots could

Fig 1. Oil reservoir with heater drawer rolled partly open.

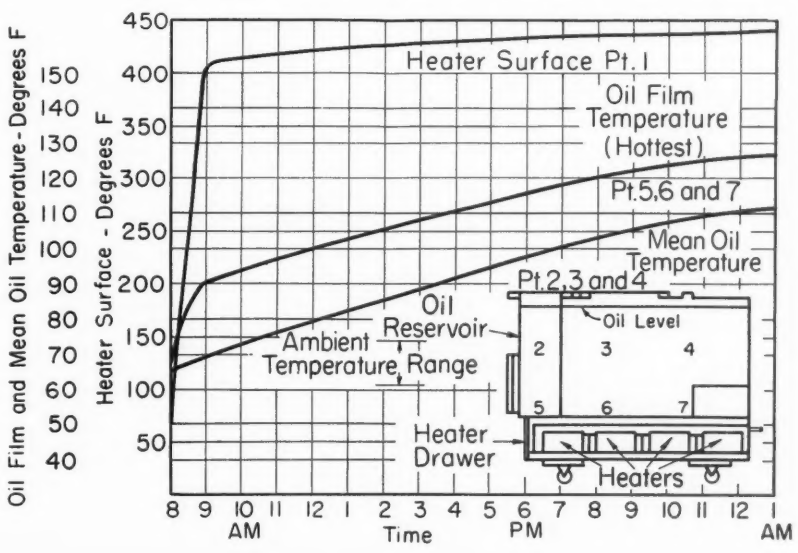

Fig. 2. Time versus temperature curve for radiant-heated oil reservoir.

*Westinghouse Electric Corp., Sunnyvale, Calif.

be ironed out to a large extent by adding a heavy plate between the strip-heaters and the tank bottom, the additional cost, weight and bulkiness make this design impractical.

However, by using a radiant-type heater which had, in itself, a large surface area, the space needed for a heater compartment was minimized while still permitting uniform heating over the full tank bottom.

The heaters selected were made from flat, heat-resistant-type glass panels one foot square. Each has a high-resistance film coated over its entire lower surface. Electric current flowing through this film causes a uniform low-temperature heating over the entire panel. Eight of these heating panels were mounted in an aluminum-lined angle iron drawer which was mounted on casters. The panels were spaced seven inches apart and seven inches from the tank bottom. Under these panels, on top of the floor plate of the drawer, a 1½-inch-thick layer of block insulation was placed. The reservoir is built on legs that are high enough to permit the heater drawer to be rolled into place beneath it.

Since twenty-four hours notice will normally be given before the wind tunnel is started, nearly all of this time can be used to bring the oil up to temperature. It was determined that four kilowatts of heater power would be sufficient, so the heaters selected were rated at one-half kilowatt each. A shop test conducted over a 17-hour period (Fig. 2) shows that at the end of the test the film temperature was 126 F., and the mean oil temperature was 111 F. The temperature differential decreased as the temperature increased, and the heater proved safe for increasing the oil temperature well above the required 110 degrees.

Current Literature

COLLOIDAL DISPERSIONS. How "dag" colloidal dispersions have been used successfully for dry-film lubrication is described in a new, illustrated bulletin, No. 438. Technical advances of recent years have demanded the improved lubricating properties provided by such solids as electric-furnace graphite, molybdenum disulfide, zinc oxide, and mica. When these lubricating solids are processed to colloidal size and dispersed in synthetic oils, glycols, mineral spirits, lactol spirits, synthetic resins and other liquid carriers, they produce desirable dry-film lubricating films which are extremely effective over a wide range of operating conditions. Various combinations of lubricating solids and carriers and their specific application in machine lubrication, metal working, and foundry operations, and other manufacturing processes are presented. (Acheson Colloids Co., 1630 Washington Ave., Port Huron, Mich.)

ELEKTRION PROCESS MANUAL. A new booklet entitled "Technical Notes on the Concentrated Oil Elektrion R" has been made available in English for American readers. It explains the effects of this very viscous oil which is created by corpuscular bombardment of a base oil when used as an additive, and discusses in detail how Elektrion R concentrates function as a viscosity index improver, pour point depressant, oiliness improver, sludge inhibitor, etc. (Michel Roegiers, DeCavel & Roegiers, S. A., Coupure 221, Ghent, Belgium)

MOLYKOTE LUBRICANTS. New concepts of extreme pressure or boundary lubrication in which solid lubricants perform a major function are described in a new bulletin, No. 103, entitled "Fringe Area Lubrication with Molykote Lubricants." Bulletin No. 104, also available, describes in technical detail the new Molykote microsize lubricant, and features a graphic picturization of the relative particle size of molybdenum disulfide powders in relation to their ability to penetrate the crevices of ground bearing surfaces. (The Alpha Corp., 179 Hamilton Ave., Greenwich, Conn.)

INSTRUMENT DATA. Applications of the redesigned ramsbottom carbon residue apparatus, an instrument for the determination of the amount of carbon residue left after evaporation and pyrolysis of an oil, are fully discussed in a new data sheet, No. 11314. Data Sheet No. 11470, also available, describes the newly developed Visi-Watt, a distillation watt-meter with a light for easier graduate reading, and Data Sheet No. 11490 covers the applications of the Mil-Shell roll grease tester, an apparatus recently designed to measure the tendency of lubricating greases to liquefy while in service. (Precision Scientific Co., 3737 W. Cortland St., Chicago 47, Ill.)

ORGANIC CHEMICALS. Prepared annually as a handy guide for chemists, engineers, purchasing agents, and laboratory workers, the 1955 booklet edition of "Physical Properties of Synthetic Organic Chemicals" presents the latest data on more than 350 organic chemicals (featuring 46 new products available from Carbide & Carbon). For easy reference, the chemicals are arranged by family groups with condensed application data; physical properties are given in tabular form. (Carbide & Carbon Chemicals Co., 30 E. 42nd St., New York 17, N. Y.)

FRONT WHEEL BEARINGS. Plaguing the average motorist has been front wheel bearing failure. "Recommended Practices for Lubricating Automotive Front Wheel Bearings," a new manual published by the National Lubricating Grease Institute, culminates the results of three years of effort involving an estimated 200

(Continued on Page 56)

HYDRODYNAMIC LUBRICATION OF A CAM & A CAM FOLLOWER

by R. Davies*

The lubrication of valve-lifting cams in an internal combustion engine is inherently more difficult than the lubrication of journal bearings because the geometry is less favorable for the formation of a lubricant wedge. Mathematical equations are derived for calculating minimum oil thickness and maximum oil pressure between a cam and a cam follower. An example is given showing the application of the equations.

Fig. 1 shows the arrangement of the valve gear of a typical 6-cylinder valve-in-head engine. As the camshaft rotates, the cam lobe pushes up the cam follower, raising the push rod and rotating the rocker arm to open the valve. The valve spring, acting through the rocker arm and push rod, holds the follower in contact with the cam. Because the cam slides on the follower, some lubrication must be provided between the two. The lubrication of such a cam and follower is inherently more difficult than the lubrication of a journal bearing. The geometry is less favorable for hydrodynamic lubrication.

The equations derived in this paper may be used to determine (a) whether the oil film is so thin that metal-to-metal contact is probable and (b) whether the oil pressure is so great that yielding of the cam or follower is probable.

The pressure gradient in the oil film is less along the axis of the camshaft than in the direction of cam rotation for films that are approximately as thick as the height of the surface irregularities on the cam and follower. For this reason, there is relatively little side leakage from these thin films, and such side leakage has been neglected in deriving Equation (8). Therefore, Equation (8) may be used as a first approximation to determine whether metal-to-metal contact is probable even though the oil-film thickness computed from it may be in error for thicker films.

Likewise, the cam and follower do not deform significantly at pressures below their yield points, and such deformation has been neglected in deriving Equation (9). Therefore, Equation (9) may be used as a first approximation to determine whether the yield points of the cam or follower are exceeded even though higher pressures computed from the equation may be in error.

*Research Laboratories Div., General Motors Corp., Detroit, Mich.

This paper was contributed by ASLE, and presented at the 1st Annual ASLE-ASME Lubrication Conference, Baltimore, Maryland, October 18-20, 1954.

The Differential Equations & Their Integration. When the Navier-Stokes equations are applied to a thin oil film without side leakage, they can be reduced to the form

$$\partial p/\partial x = \mu \partial^2 u/\partial y^2$$
$$\partial p/\partial y = 0$$
$$0 = \partial u/\partial x + \partial v/\partial y$$

Corresponding boundary conditions are

$$u(x,0) = U_o, \qquad u(x,h) = U_h$$
$$v(x,0) = V_o, \qquad v(x,h) = V_h$$

For these conditions, the pressure, $p(x)$, can be obtained by the integration of the expression

$$d[(h^3/12\mu)(dp/dx)]/dx = -d[h(U_h - U_o)/2]/dx + hdU_h/dx + V_h - V_o \qquad (1)$$

In the case of a cam and follower, the axes can be selected as shown in Fig. 2. The cam profile can be approximated by a parabola. The oil-film thickness then becomes

$$h = H + x^2/2r$$

in which H is the minimum film thickness and r is the radius of curvature of the cam at the point of minimum film thickness. If the follower does not move in the x-z plane, and if the surface velocity of the cam is U, then

$$U_o = 0, \qquad U_h = U$$
$$V_o = 0, \qquad V_h = Ux/r$$

For these boundary values, Equation (1) can be integrated to give

$$p(x) = 3\mu(C/H - U/2)x/(H + x^2/2r)^2 + 3\mu\sqrt{3C/2H^3 + U/4H}\,[x/(H + x^2/2r) + \sqrt{2r/H}\ arc\ tan\ (x/\sqrt{2Hr})] + K \qquad (2)$$

in which C and K are unknown constants of integration. The value of H is also unknown.

The Evaluation of C, K, & H. If the velocity U is positive, the cam and oil are moving across the follower from the negative x-direction to the positive. Thus the oil in the negative x-direction has not yet passed between the cam and follower, and, if sufficiently far away, the oil is at atmospheric pressure. Consequently,

$$\lim_{x \to -\infty} p(x) = 0$$

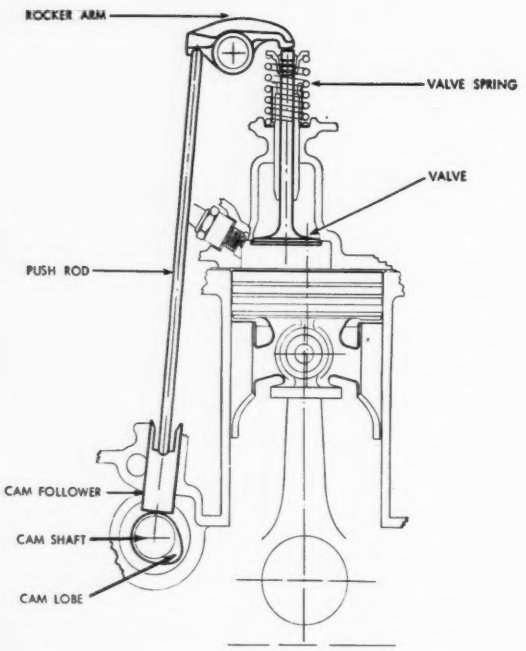

Fig. 1 Diagram of valve-gear arrangement.

For the condition to be met, the constant K must be

$$K = 3\mu(3C/2H^2 + U/4H)\sqrt{2r/H}\pi/2$$

and Equation (2) becomes

$$p(x) = 3\mu(C/H - U/2)x/(H + x^2/2r)^2 + 3\mu(3C/2H^2 + U/4H) \{x/(H + x^2/2r) + \sqrt{2r/H}[\pi/2 + arc\ tan\ (x/\sqrt{2Hr})]\} \quad (3)$$

As the oil is carried by the cam through the narrow space between the cam and follower, the pressure rises to a maximum and then decreases. Eventually, after the oil passes the region of minimum film thickness, the pressure drops to zero, but cavitation keeps it from becoming negative. Consequently, for some value x_o at which the pressure first becomes zero,

$$p(x_o) = 0, \qquad |\partial p/\partial x|_{x_o} = 0$$

and the pressure remains zero for all larger values of x. These boundary conditions cannot be used to evaluate the remaining unknown constants, C and H, explicity. However, C can be replaced by a function of the unknown x_o, since

$$\partial p/\partial x = (3\mu Ux^2/r + 12\mu C)/(H + x^2/2r)^3 \quad (4)$$

must vanish for $x = x_o$; that is

$$C = -Ux_o^2/4r \quad (5)$$

Equation (3) then becomes

$$p(x) = -(3\mu Ux/2H)(H + x_o^2/2r)/(H + x^2/2x)^2 + (3\mu U/4H)(1 - 3x_o^2/2Hr) \{x/(H + x^2/2r) + \sqrt{2r/H}[\pi/2 + arc\ tan\ (x/\sqrt{2Hr})]\} \quad (6)$$

The pressure $p(x)$ given by Equation (6) meets the condition that $|\partial p/\partial x|_{x_o}$ is zero. If a new variable y is defined by

$$y = x_o/\sqrt{2Hr}$$

the additional condition that $p(x_o)$ itself vanishes can be reduced to

$$-y - 3y^3 + (1 - 3y^2)(1 + y^2)(\pi/2 + arc\ tan\ y) = 0$$

The only real root is

$$y = x_o/\sqrt{2Hr} = 0.475130 \quad (7)$$

Both x_o and H are unknown, but if either one is somehow evaluated and the other one is determined from it by Equation (7), then $p(x_o)$ is zero.

One more condition can be imposed to determine H: The pressure must be able to support the force F, between the cam and the follower. Thus, if a is the length of the contact between the cam and follower

$$F = a\int_{-\infty}^{x_o} p(x)\,dx$$

in which $p(x)$ is given by Equation (6). Upon integration

$$F = (3\mu Ura/2H)[1 + (1 - 3x_o^2/2Hr)\{1 + x_o/\sqrt{2Hr}[\pi/2 + arc\ tan\ (x_o/\sqrt{2Hr})]\}]$$

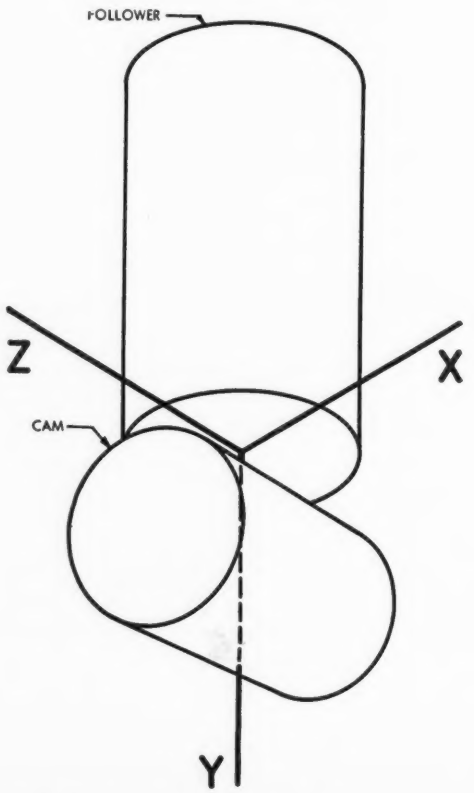

Fig. 2. Orientation of co-ordinate axes.

Equation (7) can be used to evaluate $x_o/\sqrt{2Hr}$. Then, if one solves for H, the minimum film thickness

$$Thickness_{min} = H = 2.44748\mu Ura/F \qquad (8)$$

The Maximum Pressure. Equations (4) and (5) imply that $+x_o$ and $-x_o$ are the only finite zeros of $\partial p/\partial x$. Since $\partial p/\partial x$ is negative between them and positive elsewhere, $p(x)$ is a maximum for whichever of $+x_o$ and $-x_o$ is negative and is a minimum for whichever is positive. The maximum is $p(-x_o)$ because Equation (7) shows that $+x_o$ is positive. From Equation (6)

$$p(-x_o) = 3\mu Ux_o/2H(H + x_o^2/2r) - (3\mu U/4H)(1 - 3x_o^2/2Hr)$$
$$\{x_o/(H + x_o^2/2r) + \sqrt{2r/H}[arc\ tan\ (x_o/\sqrt{2Hr}) - \pi/2]\}$$

Equations (7) and (8) can be used to eliminate x_o and H

$$Pressure_{max} = p(-x_o) = 0.280879(F^3/\mu Ur^2a^2)^{1/4} \qquad (9)$$

Example: Typical values for the valve gear of a modern high-speed internal-combustion engine are

$$\mu = 6 \times 10^{-6}\ lb\text{-}sec/in.^2$$
$$U = 100\ ips$$
$$r = 0.25\ in.$$
$$a = 0.5\ in.$$

The spring force between the cam and follower is 250 lb. However, the mass of the valve gear, which may be taken as 1.5 lb, is accelerating at 6000 in/sec^2 in such a direction that an inertia force of 23.3 lb opposes the spring force. Thus the net force is

$$F = 226.7\ lb$$

Equations (8) and (9)then give

$$Thickness_{min} = 8.097 \times 10^{-7}\ in.$$
$$Pressure_{max} = 4.428 \times 10^5\ psi$$

In this example, the pressure does not exceed the yield point of the metal, but the oil film is so thin that metal-to-metal contact may occur. In such an event, the oil must have some ability to lubricate under boundary conditions.

Patent Abstracts

Process for Preparing Pour Depressants, Patent No. 2,688,643, by R. M. Dean, E. P. Cashman & J. E. Thomas, assignors to Standard Oil Development Co.

An improved process for the preparation of alkylated aromatic materials having utility as pour point depressants which comprises alkylation of an aromatic hydrocarbon with a halogenated aliphatic hydrocarbon in the presence of about 15 to about 30 parts by weight based on 100 parts by weight of the halogenated aliphatic hydrocarbon of a hydrocarbon oil having a viscosity at 210 F. of about 140 to 200 S. U. S.

Lubricating Oil Compositions of Mixed Diester Dithiophosphates, Patent No. 2,689,220, by P. K. Mulvany, assignor to California Research Corp.

A lubricant consisting essentially of a major proportion of an oil of lubricating viscosity and from 0.05% to 2.0% of zinc salts of mixed esters of dithiophosphoric acids produced by reacting phosphorus pentasulfide with a blend of 2 aliphatic monohydric alcohols at temperatures from 120 to 200 F., one of said alcohols containing not more than 4 carbon atoms, the other of said alcohols containing from 6 to 18 carbon atoms, the ratio of the lower molecular weight alcohol to the higher molecular weight alcohol being from 1.0 to 5.0

Method of Purifying, Concentrating, and Converting Petroleum Sulfonates, Patent No. 2,689,221, by U. B. Bray, assignor to Bray Oil Co.

The process of treating a material consisting of a major proportion of hydrocarbon oil containing alkali metal mahogany sulfonates, alkali metal green acid sulfonates and water-soluble inorganic sulfate and sulfite, which comprises: forming a mixture consisting essentially of said hydrocarbon oil, said alkali metal mahogany sulfonates, said alkali metal green acid sulfonates, said water-soluble inorganic sulfate and sulfite, at least 10 parts by volume of water per 100 parts by volume of said hydrocarbon oil and sulfonates and at least 5 parts by volume per 100 parts by volume of said hydrocarbon oil and sulfonates of a hydrocarbon oil-soluble emulsion breaking liquid selected from the group consisting of butyl, amyl and hexyl alcohols, the amounts of said water and said emulsion breaking liquid being sufficient to produce three separable phases, a concentrated mahogany sulfonate phase containing hydrocarbon oil, water and emulsion breaking liquid, a hydrocarbon oil phase rejected from said concentrated mahogany sulfonate phase and an aqueous phase containing alkali metal green acid sulfonates and water-soluble inorganic sulfate and sulfite; and separating said three phases from each other.

Lubricating Oil Composition Containing the Neutralized Reaction Product of a Phosphorus Sulfide and Sugar Cane Oil, Patent No. 2,689,827, by E. O. Hook & D. W. Jayne, Jr., assignors to American Cyanamid Co.

A composition suitable for use as an improvement agent for hydrocarbon lubricating oils operative to assist in depressing the pour point of said hydrocarbon lubricating oil and preventing the deposition of hard deposits due to sludge formation in the oil comprising as its essential ingredient a neutralized reaction product resulting from chemically reacting a phosphorus sulfide with a member of the group consisting of sugar cane oil, partially saponified sugar cane oil and completely saponified sugar cane oil, and then neutralizing the product of such reaction with a member of the group consisting of metal oxides and hydroxides, said sugar cane oil having a saponification number of approximately 156, an iodine number of approximately 85.8 and an acid number of approximately 24.6.

Mineral Oil Compositions, Patent No. 2,689,828, by H. G. Smith, T. L. Cantrell & J. G. Peters, assignors to Gulf Oil Corp.

A mineral oil composition comprising a major amount of a mineral oil and a minor amount, sufficient to confer corrosion-inhibiting properties on the composition of an amino salt of an aliphatic half ester of an acid selected from the group consisting of phthalic and alkyl substituted phthalic acids, the aliphatic residue of said ester containing at least 8 carbon atoms.

Terpene-Phosphorous Pentasulfide-Sulfur Lubricating Oil Additive, Patent No. 2,689,846, by L. C. Beegle, assignor to American Cyanamid Co.

A method of preparing a hydrocarbon lubricating oil antioxidant which comprises simultaneously reacting together a bicyclic terpene, phosphorous

(Continued on Page 60)

CAGES FOR HIGH SPEED CYLINDRICAL ROLLER BEARINGS

by W. J. Anderson & Z. N. Nemeth*

In an investigation of four outer-race-riding cage-type bearings (two with leaded brass and two with nodular-iron cages) heavy wear was found to accompany cage slip. Nodular iron seemed to promote cage slip at DN values in excess of 1.2 x 10^6; consequently, bearings with leaded-brass cages showed less wear than did bearings with nodular-iron cages at very high speeds. Studies of four experimental and two conventional design roller bearings revealed that the experimental bearings operated at lower temperatures and to higher limiting DN values than did the conventional bearings. The experimental cages were designed to offer less resistance to the flow of lubricant into and out of the interior recesses of the bearing.

The importance of the cage or retainer as a component part of rolling-contact bearings has been brought to light by the fact that cage failures have ranked high among the causes of failure in high-speed roller bearings[1,2]. In Ref. 1 it is reported that the greatest number of bearing failures occurs as a result of an inner-race-to-cage seizure in roller bearings or in a ball-to-cage seizure in ball bearings. In Refs. 3 & 4 it is reported that bearing life is being limited by wear (cage, ball or roller, and race) rather than fatigue. In Ref. 5 one of the discussers states that practically all investigators have concluded that the critical factors at higher speeds are cage failure, heat dissipation, and lubrication. Such statements are rather strong, and they must not be interpreted to mean that the problem of fatigue does not exist in high-speed rolling contact bearings. They do, however, point out the critical nature of the problem. Published service experience on turbojet engines[6] indicates that failure of main rotor bearings has been one of the principal causes of engine removal and that the bearing failures have indicated the need, among other things, for more research on cage problems.

Very little information on cage design and material research for turbojet engine bearings is available in the literature. Wear and friction studies on cage materials are reported in Refs. 7 to 9.

The ultimate goal of cage research should be a cage that operates with hydrodynamic lubrication at all points of contact between it and the races and rolling elements, for then failures due to wear, galling and metal transfer would be minimized. This goal, however, appears to be impossible to achieve because the cage is supported on a sleeve bearing with an extremely low length-diameter ratio and an excessively large diametral clearance necessitated by thermal considerations (Gurney, Ref. 10). In this regard, however, partial success would be a distinct step forward.

The experimental results reported herein are based on the work reported in Refs. 11 & 12 and were conducted at the NACA Lewis Flight Propulsion Laboratory, Cleveland, Ohio. Nine test bearings are compared with respect to operating temperature, limiting DN value and wear.

TEST EQUIPMENT. The bearing rig used for this investigation is described fully in Ref. 13. Briefly the test bearing was mounted on one end of the test shaft, which was supported in cantilever fashion so that component parts of the bearing, and the lubricant flow could be observed during operation. Radial load was applied to the test bearing by means of a lever and dead-weight system.

The drive equipment, lubrication system and method of temperature measurement are described fully in Ref. 13.

TEST BEARINGS. Nine test bearings (size 215) were used for this investigation. The overall dimensions of all bearings were bore, 75 mm outside diameter, 130 mm; and width, 25 mm. Schematic drawings of all test bearings are shown in Fig. 1. Data for bearings A through F were used in the cage-design evaluation and data for bearings F through I were used for the cage-materials evaluation. Bearings A through G had nodular-iron cages while bearing H and I had leaded-brass cages. Bearings A, B, C, D, and E all had double flanged inner races (inner-race-guided rollers) and straight-through outer races. Bearings F, G, H and I all had double flanged outer races (outer-race-guided rollers) and a straight-through inner race. Bearing A had a conventional one-piece inner-race-riding cage with broached pockets, Fig. 1(a). Bearing B had a two-piece cage with fitted pockets, Fig. 1(b), and bearing C, a one-piece cage with broached pockets, Fig. 1(c), both piloted on the outer race outside the roller track. Bearing D had a one-piece cage with broached pockets, Fig. 1(d), and bearing E a two-piece cage with fitted pockets, Fig. 1(e), both piloted on the outer race in the roller track. Bearings F, G, H and I all had conventional one-piece cages with broached pockets piloted on the shoulders of the outer race, Fig. 1(f).

*National Advisory Committee for Aeronautics, Lewis Flight Propulsion Laboratory, Cleveland, Ohio.

This paper was contributed by ASLE, and presented at the 1st Annual ASLE-ASME Lubrication Conference, Baltimore, Maryland, October 18-20, 1954.

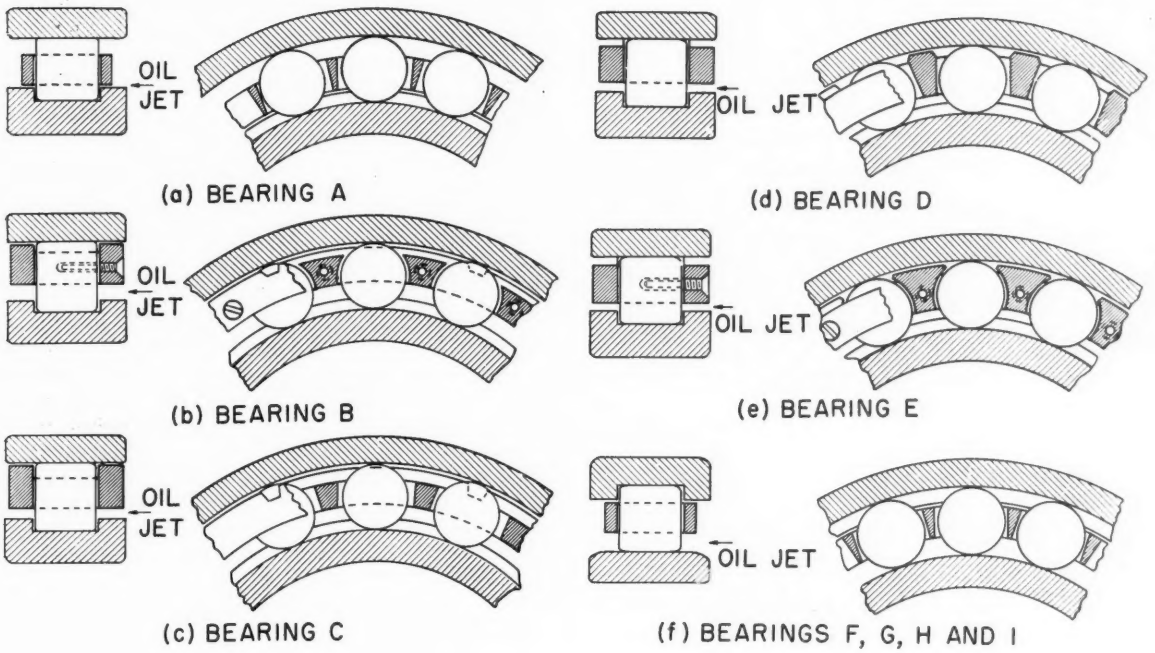

Fig. 1. Schematic drawings of test bearings.

PROCEDURE. Each test bearing was first subjected to a number of tests, collectively termed the low-speed tests, at DN values of 0.3×10^6 to 1.2×10^6, loads of 7 to 1613 lb. and oil flows of 2 to 8 lb. per min. Each bearing was then run to the maximum speed at which equilibrium temperatures could be obtained at oil flows of 2 to 8 lb. per min. These tests collectively were termed the high-speed tests. Bearings used for the cage-materials evaluation were removed from the rig and checked for wear by determining the weight loss of each component part after both the low-and high-speed tests. Bearings used for the cage-design evaluation, however, were checked for wear only after the high-speed tests.

Each test bearing was given a complete dimensional check before and after test. Hardness measurements and surface finishes also were recorded before and after test. Care was taken to assemble test bearings with nearly uniform clearances at all critical locations. Test-bearing data are given in detail in Refs. 11 and 12.

Oil was supplied to the test bearings through a single jet having a 0.050-in.-diam. orifice at an inlet temperature of 100 F., and at pressures of 30 to 405 psi which corresponded to oil flows of 2 to 8 lb. per min. The oil used was a highly refined nonpolymer-containing petroleum-base lubricating oil used to lubricate the bearings in several current turbojet engines. The oil had a viscosity of 10 centistokes (13×10^{-7} reyns) at 100 F., and 2.6 centistokes (3.2×10^{-7} reyns) at 210 F.

EXPERIMENTAL RESULTS & ANALYSIS.

Cage Design Evaluation. The effect of DN on the outer-race-maximum temperature of bearings A, B, C, D, E, and F at oil flows of 2.75 and 8 lb. per min. is shown in Fig. 2. The experimental bearings (B, C, D, and E)operated at lower temperatures than did bearings A and F under all conditions of operation, the difference in operating temperature becoming greater with increasing DN. This indicates that the four experimental types were more effectively lubricated and cooled than were the two conventional designs. Among the four experimental types, bearings D and E (cage-locating pads between the rollers) operated at slightly lower temperatures than did bearings B and C (cage-locating surfaces outside the roller track). However, the average differ-

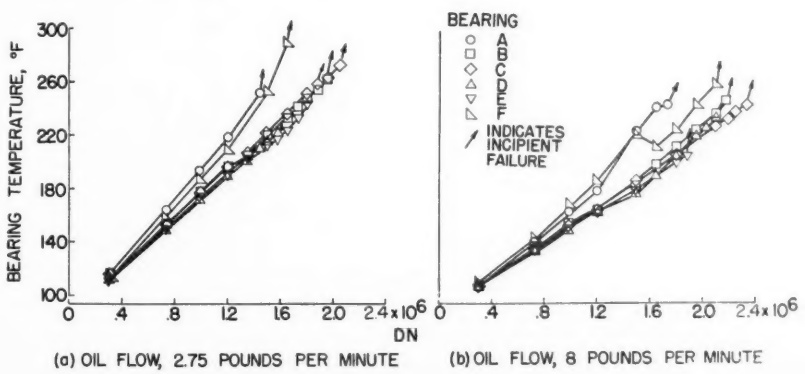

Fig. 2. Effect of DN on bearing outer-race maximum temperature at oil flows of 2.75 and 8.0 lb/min. for bearings A, B, C, D, E, and F. Load, 368 lb; oil-inlet temperature, 100° F.

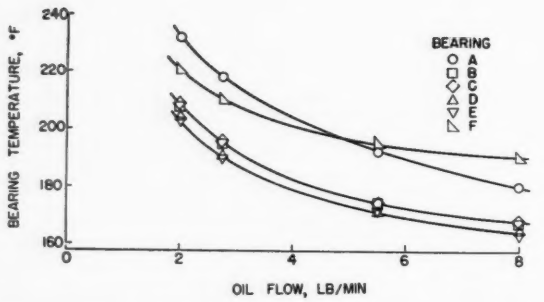

Fig. 3. Effect of oil flow on bearing outer-race maximum temperature. DN, 1.2×10^6; load, 368 lb.; oil-inlet temperature, 100° F.

ence in the operating temperatures of bearings B and C and those of bearings D and E at a DN of 1.2×10^6 was only 4 deg. F. The cages of bearings B and C have a slightly greater tendency to trap oil than do those of bearings D and E.

It is evident that bearing temperature is a generally linear function of DN for bearings A, B, C, D, E, and F at DN values of 1.2×10^6. At DN values above 1.2×10^6, however, only the curves for bearings B, C, D and E remain fairly linear. Those of bearings A and F are quite erratic and indicative of unstable operation at very high speeds. At an oil flow of 2.75 lb. per min., the slope of the curve for bearing F showed a marked tendency to increase with speed, probably because of the increasing rate of heat generation due to oil churning. Thus, the basic weakness of this design becomes more apparent at very high speeds. The curve for bearing F at an oil flow of 8 lb. per min. is quite interesting in that the bearing temperature decreased when the speed was increased from a DN of 1.5×10^6. This unusual phenomenon was caused by a sudden increase in cage slip from 1 to 2 per cent to 35 per cent and will be discussed fully.

Fig. 3 shows the outer-race maximum temperature of bearings A, B, C, D, E and F plotted against oil flow at a DN of 1.2×10^6. The curves show that bearing A operates hottest at oil flows up to about 4.8 lb. per min., while bearing F operates hottest at oil flows above 4.8 lb. per min. These results indicate that the interior surfaces of bearing A were not lubricated and cooled properly at the low oil flows and that a relatively high oil flow and a high-velocity oil jet is required for sufficient oil to penetrate through the small space between the cage and inner race. The results also indicate that, while bearing F was lubricated adequately at the lower oil flows, considerable oil entrapment between the outer-race flanges occurred at the higher oil flows and resulted in heat generation due to oil churning. It can be seen then that high operating temperatures result with the conventional inner-race-riding cage-type bearing because of its inherent difficulty of lubrication and cooling, and with the conventional outer-race-riding cage-type bearing because of its tendency to trap oil and create high churning losses. Bearings B, C, D, and E operated at lower temperatures than did the conventional bearings because the lubricant had easy access to the interior surfaces plus a low resistance path to exit from the bearing.

One of the important variables in jet lubrication of rolling contact bearings is the fraction of the oil flow which penetrates to the interior surfaces of the bearing to exit on the side opposite the oil jet. To achieve maximum lubrication and cooling this fraction should be fairly high for it is obvious that oil which merely splashes off the face of the bearing does little cooling and no lubricating. At a DN of 1.2×10^6 and 2.75 lb. per min. oil flow, this fraction was 0.18 for bearing A, 0.32 to 0.38 for bearings B, C, D and E and 0.47 for bearing F.

Each test bearing was run to its limiting DN value (that speed at which the bearing would not operate at an equilibrium temperature or at which bearing temperature rose rapidly indicating an incipient failure) at oil flows of 2, 2.75, 5.5, and 8 lb. per min. The results for bearings A, B, C, D, E and F, shown in Fig. 4(a), indicate that, in general, limiting DN values increase with increasing oil flows. The outer-race-riding cage-type bearings exhibited significantly higher limiting speeds than did bearing A. Bearing A would not run at DN values above 1.72×10^6 (23,000 rpm) whereas bearing C was operated successfully at a DN value of 2.32×10^6 (31,000 rpm). The type of cage used in bearing C seems to be best suited to ultra-high-speed operation.

Although bearing F was operated successfully at a DN value of 2.1×10^6 (28,000 rpm), operating

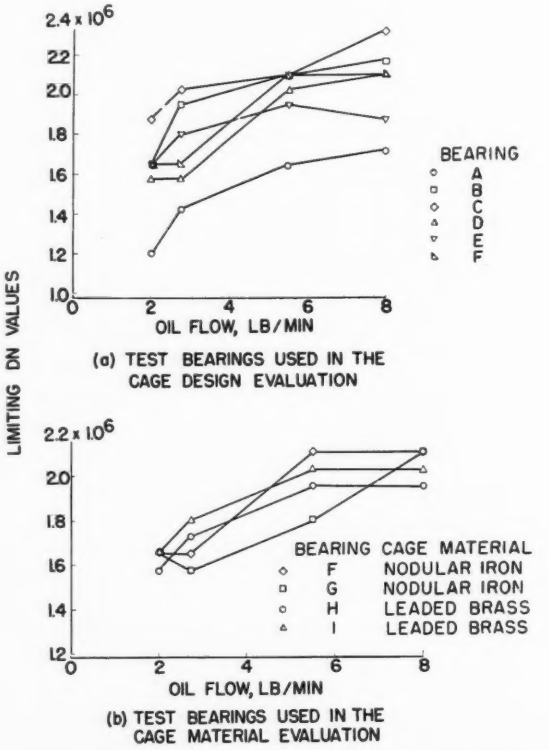

Fig. 4. Effect of oil flow on limiting DN value for all test bearings. Load, 368 lb.

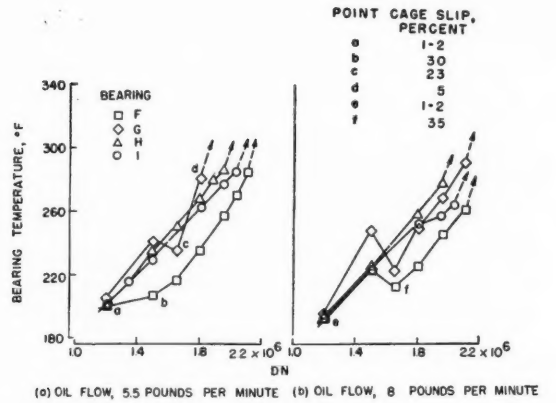

Fig. 5. Bearing outer-race maximum temperature as affected by DN values from 1.2x10⁶ to limiting value at oil flows of 5.5 and 8.0 lb/min. for bearings F, G, H, and I. Load, 368 lb.

temperatures above a DN of 1.5x10⁶ were extremely erratic because of high cage slip, which produced extremely high wear as discussed later.

Wear data for the component parts of bearings A, B, C, D, E, and F (wear indicated by weight loss) are shown in Table I. These data, together with thorough visual inspections, are invaluable because they reveal the critical or weak points in the various bearing designs. However, because wear is such a complex phenomenon the data should be used only qualitatively and not quantitatively.

Heaviest wear in bearing A occurred between the roller ends and the inner-race track flanges. In bearing A very little oil gets to this location because it is difficult for the oil to penetrate through the small space between the cage and inner race and whatever oil does pass this barrier is immediately thrown outward by centrifugal force.

Heavier wear occurred in fitted cage pockets (bearings B and E) than occurred in broached pockets (bearings C and D) presumably because of the more intimate roller contact and the consequent greater inaccessibility to lubricant of the fitted pocket. In bearing D heavy wear occurred at the cage outside periphery and the outer race inside periphery. Apparently the cage-locating pads of bearing D were not of sufficient size to support radial cage loads because very little wear occurred at this point in bearing E which had cage-locating pads that were about 60 per cent larger than those of bearing D. No severe wear occurred at any point in bearing C.

In bearing F severe wear occurred in the cage roller pockets and on the roller diameters because of the high cage slip which occurred at DN values above 1.5x10⁶ at the higher oil flows.

Cage Material Evaluation. There were no significant differences in the operating temperatures of bearings F and G (nodular-iron cages) and those of bearings H and I (leaded-brass cages) at DN values to 1.2x10⁶. At DN values from 1.2x10⁶ to 2x10⁶, however, significant variations in the operating temperatures of four test bearings were found, Fig. 5. During the course of the investigation it was found that sharp changes in the slope of the temperature-DN curve occur at the onset of cage slip. The tests seem to indicate that a roller bearing is capable of absorbing only a fixed amount of torque when operating at a specific load and oil flow. An increase in shaft speed beyond that at which the maximum bearing torque is developed will result in cage slip. Several significant points in Fig. 5 are identified, and the corresponding values of cage slip are tabulated on the figure. Percentage cage slip is defined as follows:

$$\text{Percentage cage slip} = 100\,(N_{ct} - N_c)/N_{ct}$$

where N_{ct} = theoretical cage rotational speed, rpm

N_c = actual cage rotational speed, rpm

At an oil flow of 5.5 lb. per min., the slope of the curve for bearing F decreased in going from point a to point b and at the same time cage slip increased from 2 to 30 per cent. At the same oil flow, the curve for bearing G exhibited a marked increase of slope in going from point c to point d; this change was accompanied by a decrease in cage slip from 23 to 5 per cent. Sharp changes in slope apparently indicate changes in cage slip.

The effect of oil flow on the limiting DN values for bearings F, G, H, and I is shown in Fig. 4(b). Cage-material differences produced no significant differences in the limiting speeds to which these test bearings could be run. Thus it can be seen that there were no significant differences in the operating characteristics of bearings F, G, H, and I other than differences in operating temperature at high speeds caused by cage slip.

Wear data for bearings F, G, H, and I are tabulated in Table II. Data for the low-speed tests show that there is little difference in wear characteristics between the leaded brass and nodular iron investigated at DN values to 1.2x10⁶ (16,000 rpm). The nodular-iron cages did show greater wear, but the wear of all test-bearing component parts was low and not indicative of any severe bearing-life limitations at speeds to 16,000 rpm. At 16,000 rpm and zero cage slip, the sliding velocity is 7970 fpm at the cage-locating surface and 7310 fpm in the cage pockets. At DN values above 1.2x10⁶, however, bearings with leaded-brass cages incurred much less wear than did the bearings with nodular-iron cages. Cage, roller, and outer-race wear were severe in both

Table I. Test-bearing-wear data (measured by weight loss) for cage design evaluation.

Bearing	Outer race, mg	Inner race, mg	Cage, mg	Rollers, mg/roller
A	2	504	408	19.4
B	46	324	700	42.6
C	15	259	327	5.4
D	905	195	1570	8.0
E	134	191	842	6.9
F	265	30	6270	88.6

bearings F and G. Roller wear was heavy in bearing H but this was later found to be due to a slight load misalignment. The most significant indication of the high-speed wear data is the relative wear of the four test-bearing cages. Comparison of the high-speed DN curves, Fig. 5, and the high-speed-test wear data, Table II (b), suggests strongly that cage slip is responsible, at least in part, for high wear rates and is therefore highly detrimental to bearing life.

Cage Slip Tests. The exact effects of increased cage slip, other than the tendency to reduce bearing torque and bearing operating temperatures because of reduced sliding velocities within the bearing, were unknown at the time of the cage material investigation so special tests were run to determine the effect of cage slip on the wear rates of the bearing component parts. Three tests, each of 4 hr. duration, were run at a DN of 1.2×10^6 and at an oil flow of 2.75 lb. per min. Bearing I was used because it had sustained the least wear. The load during each test was different to vary the cage slip. The results of these tests are tabulated in Table III. Bearing I incurred no appreciable wear when cage slip was as high as 30 per cent but very severe wear occurred at 95 per cent cage slip. At DN values of 1.5×10^6 and higher and at a load of 368 lb., bearings F and G sustained very severe wear at values of cage slip on the order of 30 per cent, Table II (b) and Fig. 5. Cage material undoubtedly has an effect on wear rate, but for both cage materials investigated the results seem to indicate that additional important factors affecting wear are the sliding velocities and the loads between the component parts. The sliding velocity between the rollers and inner race is, in turn, a function of the percentage cage slip and the DN value.

Table II. Test-bearing-wear data (measured by weight loss) for two cage materials.

(a) Low-speed tests Bearing	F	G	H	I
Cage material	Nodular iron	Nodular iron	Leaded brass	Leaded brass
Outer-race wear, mg	17	23	16	5
Inner-race wear, mg	5	1	7	2
Roller wear, mg	28	67	68	22
Cage wear, mg	37	51	18	13

(b) High-speed tests Bearing	F	G	H	I
Cage material	Nodular iron	Nodular iron	Leaded brass	Leaded brass
Outer-race wear, mg	248	515	69	4
Inner-race wear, mg	25	34	23	10
Roller wear, mg	1744	1882	883	48
Cage wear, mg	6233	8913	30	18

It can be concluded then that cage slip, despite the fact that it is accompanied by a decrease in heat generation, is extremely harmful, from a wear standpoint, to a cylindrical roller bearing of the type studied. For this reason nodular iron cannot be recommended as a cage material for this type bearing because it apparently tends to promote cage slip. It seems most probable that, with nodular-iron cages under poor lubrication conditions introduced by high sliding speeds, the friction forces between the cage and rollers and between the cage and outer race become great enough to exceed the tractive force between the rollers and inner race.

SUMMARY. Two experimental investigations, one to evaluate the high-speed performance of six different roller-bearing cage designs (four experimental and two conventional) and one to compare the merits of leaded brass and nodular iron as roller-bearing cage materials are reported. Nine 75-mm-bore cylindrical roller test bearings were operated over a range of DN values from 0.3×10^6 to 2.3×10^6, radial loads from 7 to 1613 lb., and oil flows from 2 to 8 lb. per min. with a single-jet circulatory oil feed.

The four experimental bearings used in the cage-design investigation had outer-race riding cages and inner-race guided rollers. This design was found to give the best over-all performance based on limiting DN values and bearing temperatures. The better performance of this type bearing over both the conventional inner-race-riding cage type and the conventional outer-race-riding cage type with outer-race-guided rollers is a result of the relative ease of lubrication and cooling and of the adequate oil exiting paths which minimize oil entrapment and churning losses.

The conventional inner-race-riding cage-type bearing operated hotter than the four experimental bearings throughout the range of speeds and oil flows investigated because it is inherently difficult to lubricate and cool. The conventional outer-race-riding cage-type bearing was found to be lubricated adequately at relatively low oil flows (2 and 2.75 lb. per min.) but at oil flows of 5.5 and 8 lb. per min. this bearing operated hottest because of excessive churning losses.

Cage-pocket type (broached or fitted) had little or no effect on bearing operating temperature, but cages with fitted pockets incurred greater wear in the roller pockets.

Table III. Test-bearing-I wear data for cage-slip investigation (measured by weight loss). Conditions for each test: time, four hr; DN, 1.2×10^6 (16,000 rpm); oil flow, 2.75 lb/min; oil-inlet temperature, 100° F.

Test	Load, lb.	Average cage slip, percent	Wear, mg			
			Outer race	Inner race	Rollers	Cage
1	368	2	3	-1	3	6
2	113	30	4	10	3	6
3	7	95	7	132	1762	49

Data for four conventional outer-race-riding cage-type bearings (two with leaded-brass and two with nodular-iron cages) were used to compare these cage materials. The results indicate that nodular iron has a tendency to promote cage slip at high speeds (DN values above 1.2×10^6), and, further, that cage slip is very harmful, from a wear standpoint, to the type of bearing used. Consequently, bearings with nodular-iron cages showed more wear of rollers and cages (especially in cage-roller pockets) than did bearings with leaded-brass cages. Purposely induced cage slip (at light loads) in a bearing with a leaded-brass cage produced heavy roller wear and light cage wear. This suggests that cage material may be a factor in reducing cage wear in a bearing operating under slip conditions. At speeds below 16,000 rpm wear in the four bearings used in the cage-material evaluation was light and was not of a magnitude that would indicate a very limited life.

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Coming Events

FEBRUARY

18-19, National Society of Professional Engineers (annual Spring meeting), Hotel Charlotte, Charlotte, N. C.

MARCH

20-23, American Institute of Chemical Engineers, Kentucky Hotel, Louisville, Ky.

28 to April 1, American Society for Metals (9th Western exposition and congress), Pan American Auditorium, Los Angeles, Calif.

APRIL

13-15, American Society of Lubrication Engineers (10th Annual Meeting & Exhibit), Hotel Sherman, Chicago, Ill.

13-15, National Petroleum Association (52nd semiannual meeting), Cleveland Hotel, Cleveland, Ohio.

18-20, American Society of Mechanical Engineers (Spring meeting), Lord Baltimore Hotel, Baltimore, Md.

MAY

6, American Association of Spectrographers (6th annual conference; subject: "Industrial Application of Spectroscopy"), Chicago, Ill.

JUNE

7-10, American Welding Society (annual Spring technical meeting and welding show), Municipal Auditorium, Kansas City, Mo.

SEPTEMBER

12-16, Instrument Society of America (10th annual conference and exhibit), Shrine Exposition Hall & Auditorium, Los Angeles, Calif.

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LUBRICATION APPLICATION PROBLEMS IN COAL MINING (Panel)

G. R. Spindler, Panel Chrmn.*

The coal mining industry presents lubrication application problems which are peculiar to it and are the result of the damp, dusty and cramped conditions under which the mining equipment must operate. The advent of complex machines and increased economic pressures have intensified the problems. This condensed panel report summarizes opinions of experts in this field.

G. R. SPINDLER, West Virginia University: It is traditional in mining that the machinery must operate under severe conditions which are conducive to wear. For this reason the maintenance of mining machinery in the coal industry is carefully planned to obtain reasonable continuity of performance and low cost of production. It is a procedure that currently is exciting more interest and concern on the part of mine owners and mine managers than almost any other administrative problem involved in the actual production of coal.

The coal mining industry is faced with the dual problem of rising costs, in so far as production is concerned, combined with reduced realization and market for the product. It is difficult to estimate the average decrease in return per ton of coal marketed during the past eighteen months to two years, since the figure varies considerably from field to field and with individual mining operators. The fact remains that the market value of coal has declined substantially during the recent past. In one case, two base reductions in the price of northern West Virginia coal during the past eight months have amounted to fifty cents per ton. Competition for markets certainly will have reduced prices still further for certain individual producers. As additional complications there have been the increased costs that come as the result of part-time operation, high equipment prices, and the difficulties that accompany intermittent performance of crews and machines. Coal has been hard pressed in certain market areas by competitive fuels and it is possible that even further price reduction may be necessary to recover or even retain its market potential.

*West Virginia University, School of Mines, Morgantown, West Virginia.

This panel was sponsored by the ASLE Technical Committee on Lubrication Equipment, and presented at the ASLE 9th Annual Meeting, Cincinnati, April 6, 1954.

The industry also has entered into a period of changed mechanization of the production processes that bring into being more expensive, more complicated machines with the inevitable result that productive capacity is concentrated into fewer units. The matter of maintaining continuous and efficient operation of such machines is of more and more importance as the trend progresses.

W. R. WOOD, The Berwind-White Coal Co.: As a beginning it is pertinent to point out some of the salient features of coal mines affecting the maintenance and lubrication of machinery used in the mines. In the mines with which I am connected, located in Central Pennsylvania, as an example, coal is mined in the Miller or B Seam and in the Upper Kittaning or C Seam. Seam thickness is roughly 35 to 45 inches in the B Seam and 48 to 54 inches in the C Seam. Physical conditions in these mines result in a saturated atmosphere at about 57 degrees Fahrenheit. Moist to wet surfaces are usual. In the producing areas coal dust is always present, even though in most places water sprays are used to allay the air-borne dust particles. In addition to the inevitable coal dust, rock dust is intentionally air-blasted and hand-dusted onto all surfaces to reduce or eliminate the possibility of coal-dust explosions.

Machinery used in these surroundings is subject to moisture condensation and consequent rusting. Dust works into every crevice and adheres to every open surface. In any place on a machine where dust and oil or water can pack, they do so and the mixture often becomes hard and dense. The mechanical equipment in these mines includes Joy miners, loading machines and shuttle cars, cutting machines and tramming trucks, belt conveyors, chain conveyors, car pullers, locomotives and various types of pumps, all powered by direct-current motors. Older machines were built with little attempt to exclude dust from the gears and bearings. More recent designs incorporate, in varying degree, seals and cases to keep dust out of the working parts. Most modern machines incorporate hydraulic equipment.

The nature of mechanized mining demands rugged machinery. Gears, bearings, and frames are often called upon to withstand strains, shock loads

and abuse that few other types of machines ever encounter.

In mechanized mining each machine must be kept in good running order to maintain production. This requires a regular maintenance procedure. Due to the normal difficulties and expense of moving heavy mining machines, they are usually kept as much as possible in the working place. It is there, near the working face, that regular daily maintenance is done. Replacement of worn or damaged parts is difficult and expensive. The loss of production resulting from failure of a machine during the regular working shift can make the operation of any machine in the mine extremely unprofitable.

Lack of proper lubrication can be and often is a cause of worn bearings, seals and gears. Worn or improperly adjusted bearings and gears greatly amplify the stresses due to shock loads and in some cases actually produce conditions resulting in shock load. These increased stresses often cause premature failure of machine parts and frequently result in motor failures. Good maintenance depends to great extent upon care, knowledge and integrity of the maintenance personnel. Close and continual supervision is seldom practicable.

Difficulties in effecting good lubrication are mostly due to the general conditions described. In addition, there are several problems that involve the kinds of lubricant, methods of handling and transporting and the application to the machines. There are so many types of bearings and gears, that to select an ideal lubricant for each would require a very considerable assortment of oils and greases. Manufacturers' recommendations are sometimes confusing and sometimes seem to be somewhat arbitrary.

The relationship between the price of the lubricant and the cost of repair and maintenance is not easily determined because of the prominent part played by maintenance. For example, ball or roller bearings are often completely destroyed either because of lack of lubrication or because of improper adjustment. To determine which was the primary cause of trouble is usually difficult.

To obtain satisfactory and practical mine lubrication requires consideration of several factors: (1) Minimize the grades and kinds of lubricant. (2) Choose quality to suit the most severe service conditions. (3) Avoid, if possible, lubricants that resemble each other too closely in both color and viscosity or consistency. (4) Establish regular lubrication schedules. See that every point is lubricated at proper intervals and in the proper amount and manner to eliminate or at least minimize starved bearings and entrance of dust into oil seals and bearings. (5) Keep filters and screens in place and in good condition. Check oil seals. (6) Eliminate contamination of lubricants at all storage transfer points as well as at point of application. Tight cans, transfer pumps or a mobile lubrication service station can help to accomplish this result.

At the mines to which reference is made, application of the first four points mentioned have resulted in satisfactory lubrication of continuous miners, shuttle cars, cutting machines, loading machines, and all other machines in the production areas with four lubricants: a straight mineral oil of medium price and good quality for hydraulic systems, air tools, small compressors, ring oiled bearings and squirt cans; an EP type of oil for all gear cases and waste packed bearings on cutting machines; a calcium-base grease for all grease gun fitings; and a high-grade, high-temperature grease for all ball and roller bearings on motors.

J. J. SLOMER, Goodman Manufacturing Co.: Machine design has undergone radical changes in the mining industry. Years ago, with some machines working on the face, it was necessary to lubricate the bearings about every twenty minutes. If that was not done, the bearings would run hot. Black oils and cup greases were generally used. From that original beginning, from machines which sold for around $2,500, the industry has gone to continuous mining machinery where the lubricant reservoirs hold about a barrel of oil and refill oil need be added only about once a week. The cost of these new machines is about $100,000.

Effective lubrication begins on the drawing board if lubrication is to take a minimum of the operator's time. Each year shows an improvement in this regard. Schools are now cooperating in the teaching of lubrication from a design standpoint, but we need the cooperation of the operators and management to speed progress.

The cost of the machine must be related to the cost of lubricant. Consider the design of the new miner which we recently sent to the southern part of Illinois. The master mechanic was asked to keep the hydraulic oil clean and provide for individual storage of new oil. He said he could not do that in spite of the high cost of the equipment. If the management cannot plan means to keep the hydraulic oil clean, it doesn't leave the manufacturer too much leeway.

We, as manufacturers, try to give information in our bulletins on each of the machines in order to impress and instruct the users on the importance of lubrication. We keep the number of lubricants down to a minimum; primarily four as follows: hydraulic oil, a No. 2 NLGI grease for motors, No. 00 NLGI grease for pressure lubrication, and a viscous general purpose oil (SAE 50).

G. C. DELMASO, Joy Manufacturing Co.: The machinery used to supplant the pick and shovel has advanced from machines that were entirely mechanical to units that employ hydraulic systems to operate many different operating elements. In particular, hydraulic transmission of power has been applied to coal cutters, coal loaders, shuttle cars and continuous miners. Paralleling the advance of hydraulic power transmission came the development of improved hydraulic seals, hoses, valves, cylinders, pumps and etc.

The first hydraulic system allowed excess leak-

age and, as a result, did not justify the cost of a good hydraulic fluid. Today, precision-made equipment, improved hoses and fittings, and automatic controls for uniform pressures, justify the development and use of high quality hydraulic fluids.

The hydraulic systems of coal mining machinery are basically simple. As most hydraulic systems diagrams will show, a pump draws oil from a reservoir and delivers it under pressure to a block of control valves. A small quantity of the oil is by-passed to a filter which removes contaminants and returns clean oil to the reservoir. Oil pressure is controlled by a pressure-relief valve and by-passes the oil to the pump suction. In some systems, a relief valve is provided in each circuit and is set to limit the pressure to the maximum required to operate the cylinder or hydraulic motor in the circuit.

The design of today's highly specialized mining machinery must include the method of application of the lubricants and consideration of types of lubricants available in the particular area. Many serious lubricant errors can develop by trying to use hydraulic oil for other lubricating purposes or attempting to use the heavier gear oils for the hydraulic fluid medium. Not all petroleum oils qualify as good hydraulic fluids. This being the case, let's consider the characteristics that a good hydraulic fluid should have when used in mining machinery.

First is viscosity, since the machine will not operate properly unless the viscosity is within a certain range. The choice depends almost entirely upon the design of the system. An oil of too heavy body will flow sluggishly with subsequent slow machine action. If the oil is too light it will flow too freely and machine response may be so fast that proper controls will be difficult. Moreover, internal and external leakage may be excessive. Internal leakage tends to occur between the stationary and moving parts of the pump or hydraulic motors and between the high-pressure and low-pressure side of piston and valves. External leakage can occur at the piston rod and shaft packing. Too light an oil may not protect the pump or hydraulic motor parts against wear.

Second is film strength. Only very thin lubricating films exist between pistons and their cylinders, between spool-type valves and their bodies, between pump gears and their pressure plates and between piston rod and packing. High film strength is particularly required when the hydraulic fluid also lubricates the main transmission gears under the shock loads imposed during normal operation.

Third is resistance to oxidation. Whenever oil is in intimate contact with air there is a tendency for the oil to oxidize. The rate of oil oxidation depends on the chemical stability of the oil, temperature and the amount of oil surface exposed to oxygen. As oxidation progresses, a condition will be reached when oxidation products become insoluble in the oil and form deposits in the machine. The deposits may be black, sticky, slimy material that combines with contamination to form sludge or dark-brown varnish-like film on the metal surfaces. Either type of deposits, if excessive, will cause trouble in hydraulic systems. In hydraulic systems of mining machinery, oxygen of the air always is in contact with the oil. Air is further introduced in the oil by entrainment by return-oil dropping and splashing into the reservoir, by air leaks in the pump suction, by too low an oil level, or by splashing in a gear case. The quantity of air dissolved in oil depends on pressure and temperature. The rate of oxidation increases rapidly with temperature where the hydraulic system is separate from the gear case.

A temperature increase develops as the oil flows through the pressure regulating valves. During most of the time a machine is in operation, this flow is at full pump capacity because the oil is used only to adjust the machine element position and is not used again until position has to be changed. Some hydraulic systems are arranged so that high pressure is developed only when the system is called on to do work and the temperature of the hydraulic fluid is generally lower. When the hydraulic fluid lubricates the main gears, operating temperatures may reach 200 degrees Fahrenheit because of additional heat developed by the clutches in the gear cases.

Where the hydraulic system is separate from the main gear case, the reservoir is about the only place through which contamination can enter the system despite the prevailing dust conditions. In wet mines, water vapor may condense when the machines are idle; this can cause emulsion formation and result in sluggish operation.

F. C. ROBERT, Gulf Oil Corp.: Lubrication application problems can be solved by the cooperation of (1) the coal mining equipment companies, (2) the lubricating equipment companies, (3) the oil companies, and (4) the coal companies.

(1) Coal mining equipment can and should be built with lubrication in mind. All of the points to be lubricated should be accessible, and more attention given to the use of automatic lubricating systems, zoning of the lubrication points, use of oil-impregnated plain bearings that require no further lubrication, and using total-seal ball or roller bearings for transmissions.

Total-seal bearings not only are good for preventing excess leakage in transmissions when mechanical seals are worn, but they also prevent any gear chips from getting into the bearings to destroy them.

The number of lubricants recommended for a piece of mining equipment should be kept to a minimum to prevent mixup in the lubricant application. If one oil is used the chances of its being properly applied is one hundred percent, but as the number of oils used increase, the odds drop correspondingly. In other words, if two oils are involved the chance of their being applied properly is fifty-fifty; three oils, one to three; etc. The same thing applies to greases. Chances of error in selecting a lubricant are especially great for face equipment where supervision is not adequate and the conditions are not the best.

(2) Lubricating equipment should be designed

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with the coal mines in mind. Since coal seams vary in thickness from inches to feet, a grease gun that works at one operation may be useless at another operation. For example, a grease gun weighing forty or fifty pounds presents a formidable task when it must be carried for five or six hundred feet in low coal.

In low coal there is a definite need for a small grease bucket and a portable electric grease gun, each holding around fifteen pounds of lubricant. The power gun could be plugged into a power take-off on the loading machine or some power source on other equipment, to obtain virtual service-station lubrication for face equipment.

An electric grease gun will make hand lubrication of face equipment fast and positive, facilitating onshift lubrication, especially for shuttle car and belt conveyor mining.

(3) Oil companies must continue to improve their lubricants with use at coal mines in mind. Multipurpose lubricants should be developed and improved to make simplified lubrication programs possible in line with increased use of mechanical equipment.

The lubrication engineer should be given the opportunity to suggest the lubricants that will be used on new equipment on its arrival at the mine. In this way the number of lubricants used on operating equipment can be kept to a minimum and reduce the stocking problems.

(4) Mine management should buy mining equipment with lubrication in mind, being sure the former will fit into the lubrication program and making the necessary adjustments to be ready when the equipment arrives on his property.

The lubrication program should include an oil house and steps taken to see that the people responsible for the lubricant application are supplied with the proper information and equipment.

The lubrication crews should be dependable because they can make or break all of the planning up to this point. If the men do a poor job of application, then it does not make any difference how good the mining equipment is designed, the type grease gun supplied, or the quality of the lubricant. The program from a lubrication standpoint will be a failure.

Lubrication application is so important to the operation of the equipment that the whole program of lubrication should be a management function.

C. J. MOSER, The Texas Co.: It is a well-established fact that the coal industry has mechanized itself tremendously in the last ten years. The coal operator, the coal equipment manager, and the petroleum industry have worked hand in hand to further this progress. Because of the very adverse conditions encountered in all mining operations, the coal operators have demanded simplification of lubrication. This presents a problem for the lubrication people in particular, as the different equipment manufacturers recommend a wide variety of lubricants for the different mechanisms.

As outlined by the other panel members, there are untold questions and problems on lubrication that face the coal operator, such as proper selection of products, cleaning-plant lubrication, underground equipment lubrication, proper storage and handling of products, etc.

QUESTIONS & ANSWERS

Q. What type and consistency grease is best suited for general purpose service on underground equipment?

A. That will depend on the type of seals and equipment involved. Generally, a No. 2 NLGI lime-base or lithium-base grease will be best, being resistant to water washing. Sometimes, however, a No. 1 or even a No. 0 grade will be preferable where ease of application must be considered. Where ball and roller bearings are involved with the other mechanisms, the lithium soap base grease is the best general purpose product. (Moser)

Q. Why don't you men get together and recommend the same kind of lubricants?

A. That is a very good question, and I think that we have been getting together. The only real difference we find is in some of the charts where EP lubes are called for. We have taken them off our list, because we have friction drums and friction clutches, and where EP lubes simply prevent operation. I would like to see a meeting of equipment manufacturers having as its purpose the simplification of required lubricants. We call for four lubricants in our equipment. Since that is approximately what is specified by most of the other manufacturers, I think we are already pretty close to a final solution. (Slomer)

Q. What is the average life of various underground mining machines? Also, what are the costs of the machines when new?

A. As a general rule the life depends entirely on whether the machines are single or triple-shifted. However, the maintenance programs that a lot of the operators have in effect today have done much to extend the life of these machines. Shuttle cars, for instance, have an average of about nine to ten years. Loading machines have a somewhat shorter life, probably around six years. As far as cost is concerned, the range will be from $10,000 to $30,000. Costs have gone up considerably in recent years. For instance, loading machines today have practically doubled in cost. Continuous miners are another example: they started out at about $50,000; with new developments in the next year or so, they will probably go up to about $90,000. With this increase, however, there is the advantage of greater ease in operation, more rugged construction, and longer life. (Delmaso)

Q. Is grease or oil best for gear cases?

A. In general, oil is best although where there may be leakage a semifluid grease may be advisable. Oil, however, gives more dependable lubrication because medium gear oils are fortified to carry greater loads. (Robert)

Q. Is there any practical nonflammable hydraulic fluid for underground machines?

A. None are generally available as yet. The cost factor also is a deterrent. (Delmaso)

Q. What determines the proper grade of grease for mine car wheels with tapered roller bearings?

A. Normally the requirement calls for a grease of low cost that will withstand moisture conditions, i.e., a lime-base product of No. 1 or No. 2 consistency. Opinions differ as to quality, the type of oil used in the grease depending on the lubrication period which will vary from three months to three years. (Wood)

Q. Have you found it necessary to specify a certain type or base of grease to be used in so-called life-time prelubricated bearings?

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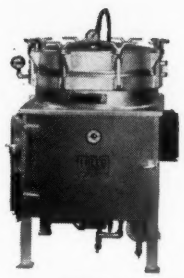

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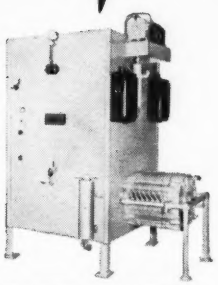

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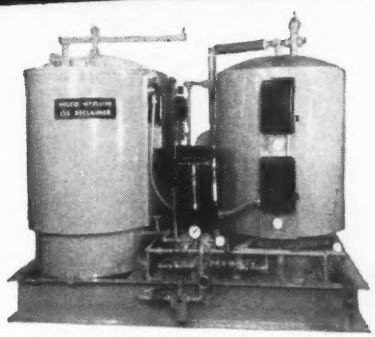

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A. I am not going to attempt to answer that question as asked. Our experience in the mines with life-time prelubricated bearings just has not worked out satisfactorily. (Wood)

Q. Is it advisable to mix oils for mining machines?

A. Mixing lubricants is not good policy. Select and use the grade of lubricant in line with the manufacturers' recommendations. (Delmaso)

Q. Are there any advantages in the use of EP Lithium over non-EP Lithium greases in the lubrication of coal mining machines?

A. Yes, it permits the use of line lubricant for all applications, and precludes the chance of mixing lubricants. For example, if the greaser were to apply a semifluid lubricant to a commutator bearing of a motor, the armature would be damaged in a very short time. Use of a multipurpose lithium-base grease would prevent this. (Robert)

Q. Does this apply to the other types of lubricants?

A. Generally, an EP-type gear oil is definitely advantageous, although it should not be used in friction clutches. The terrific heat and slipping of these clutches can cause carbon deposits on the plates in gathering head transmissions and other parts such as the wheel units that employ a worm under high loads. An extreme-pressure lead-base or even a cylinder oil with about five to eight percent compound does a better job than a straight mineral oil. (Moser)

Q. Specifically, why are not fire-resistant hydraulic oils advocated for mining machines?

A. Primarily, due to price. In other words, the use of fire-resistant oils for mining machinery becomes a matter of economics. Until the price is comparable to petroleum products, I do not think that they will be used. (Slomer)

A. It is my understanding that there are two basic types of organic compounds used: (1) water soluble, (2) synthetic organic compounds. The synthetic organic compounds have a high flash point and can't be classified as nonflammable, only fire-resistant. If an electrical short circuit develops in a cutting machine at the face and a fire is started, it is possible that the synthetic hydraulic fluid would still burn and create a toxic atmosphere that could endanger the lives of the workers. When fire-resistant water-soluble hydraulic fluids are used, it is important to be sure that the pump mechanisms are lubricated properly at the high pressures which prevail. Although the water-soluble fluids are cheaper than the synthetic organic compounds, the design of most mining equipment would have to be changed to use them successfully. So, when we consider using fire-resistant hydraulic fluids, we must first know the type product and whether it can be used on the equipment. (Robert)

Q. What kind of grease is best for motors?

A. An all-purpose lubricant is very good, although I think most of the motors that are designed for relubrication are lubricated with a soda-base type material, of usually No. 2 NLGI consistency. (Robert)

Q. Why aren't sealed ball bearings more generally used in transmissions?

A. I think that is a matter of the designer's choice. We use them on some machines such as belt rollers, but even when we use sealed-for-life bearings, the grease has to be replenished about once a year, due to the high-temperature of the motor rather than the grease and the bearing itself. While mining machine motors are usually rated at ninety degree rise for the commutator, in service I have taken readings up to 150 C. (300 F.). (Slomer)

A. I would just like to add a few remarks as to the use of seal bearings in transmissions. We have been using them very successfully in controlling the oil level. In other words, having one common transmission with a partition between them you can have one level higher than the other level through the use of these bearings. For instance, when conveyors are raised, the oil level in the gear case changes to the extent that oil might pour back to the armature bearing. Seal-type bearings have eliminated a lot of troubles that were formerly due to oil getting into the motors. (Delmaso)

Q. What type of cable lubricant furnishes the best lubrication and protection to wire rope on skip hoists under dirty and wet conditions?

A. An inhibited-type product that is fluid enough to penetrate to the center core of the wire rope, applicable without heating, able to remain pliable under all atmospheric conditions and resistant to evaporation and water. (Moser)

Q. Can I use SAE 70 cylinder oil for hydraulic oil? What factors dictate the recommended viscosity of a hydraulic oil for most underground equipment?

A. SAE 70 oils have been used on some machines. It depends upon the type of circuit and other factors. For example, a filter in the line may prevent reasonable flow of oil of this viscosity. The length of the intake line is another factor as it may cause cavitation at the pump. Usually, we don't recommend oils as heavy as SAE 70 to be used in the circulatory systems for the hydraulics. We try to keep a light lubricant or light hydraulic oil on longer lines in order to keep the pressure drop down to a minimum. Heavier oils increase the pressure drop and a lot of times they cause additional heat in the system. Where there is a limited supply in the tank this heating is objectionable. (Slomer)

A. The viscosity of oil depends also on the type of and design of pump. A Vane-type pump as a rule will require an oil not much heavier than a 300 SSU @ 100 F. In a gear-type pump an oil as heavy as SAE 70 can be used. Actually a properly-operating gear-type pump could function on an oil anywhere between 300 to 800 SSU @ 100 F. A good many of the coal operators are tending toward heavier oils because they can use the same oil in some gear cases, frictions and other places, eliminating the need for an extra product. In other words, they go to a heavier viscosity to

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secure a general-purpose hydraulic oil. However, filters must be serviced properly. A plugged filter can make a hydraulic unit very sluggish on cold starting. (Moser)

Q. Why is it necessary to use expensive oils in hydraulic equipment?

A. Due to the fine finish on the valves and pump elements, also the prevailing high pressures. Premium hydraulic oils containing foam additives and fortified to resist rust and oxidation will insure greatest protection of the costly mechanisms which make up modern hydraulic power transmission systems. (Slomer)

Q. How can grease and oil be kept clean in a coal mine? How detrimental is a little dirt in lubricants?

A. Well, lubricants can be kept clean but the ideal seldom is attained. The proper procedure is to take the oil or grease container in which it is received from the manufacturer, pumping it directly from that container into a distribution reservoir or grease applicator. Later, if the man handling the lubricator cleans off fittings before he starts to pump grease it is still clean as it goes into the machine. In service, seals play an important part in preventing contamination of lubricants. But they must be kept in good condition. (Wood)

Q Can opinions be stated as to the types of bearing seals most effective for ball and roller bearings, for instance, on mine fans?

A. The type of seal is more or less secondary to its manner of installation and adjustment. For example, if the seal in a fan bearing is too tight and excess lubricant is supplied, failure may result due to the increased heat generated by the frictional resistance of the grease. As to the best seal, I would judge that it is the one that best suits your problem plus some facility for bleeding excess grease from the bearing as the speed increases. This will prevent damage to the seal from pressure build-up and damage to the bearing from excessive heat. (Robert)

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Q. What is the trend in the industry towards centralized lubricant storage and handling facilities?

A. There are a number of large mining companies that have installed centralized systems on their equipment to good advantage. The only complaint, as voiced by some operators, is that it has to be maintained. Someone will have to check the machine daily. Just a short, visual inspection is necessary to catch a broken line or fitting and arrange for its replacement. If this is not done, of course, the bearing involved eventually will fail due to lack of lubrication. Centralized systems in tipple and cleaning plant equipment are almost universal today with every new plant that is constructed. In underground equipment, however, centralized lubrication is not as widely used. (Moser)

Q. Is it possible to list the number, types and uses of the greases that would be taken into a typical mobile loading section involving a loading machine, cutting machine, roof bolt drill, coal drill, etc.?

A. That, of course, depends on the type of equipment. You can get by using three products, other than a motor bearing grease which in most cases is applied on the outside due to the infrequency of application. The three products are: (a) A general purpose premium grade hydraulic oil somewhere between the range of a 500 and 800 SSU @ 100 F. This product to be used in all hydraulic systems where gear-type pumps are employed and also gear cases employing frictions. If a vane-type pump is employed, a 300 SSU @ 100 F. oil is necessary. (b) A lime-base grease with a high-viscosity oil of either a No. 1 or No. 2 NLGI consistency for use as a general purpose bearing grease. (c) An extreme pressure gear-type lubricant for all gear cases, except those employing friction clutches. (Moser)

Q. If sufficient concentration of equipment is in use at the face, will not a lubrication crew afford more satisfactory lubrication by giving the responsibility of lubrication to the machine operator? What is the practice?

A. Well, in the mines with which I am connected, it is the practice to have a mechanic lubricate the machine. He lives with the machines, looks after them, attends to lubrication. Other mines do have or are carrying out some experimentation, at least, with a lubricant specialist who goes around with a lubricating car. In some cases, I believe they actually have power pumps and long hoses to run up into the rooms. (Wood)

Q. What causes so much gummy material to collect in a gear case?

A. That problem is confined largely to the coal mines. You don't find it on outside machinery to any extent. In the coal mines it can be a very serious problem, especially in gathering-head gear cases. Those cases are sealed with a leveling seal and have a special grease fitting for filling the seal space. They cool off, of course, during idle time. As they cool, "breathing" occurs and air is drawn into the case unless it is completely filled with lubricant, carrying dust and moisture in with it. On starting up, the case becomes warmer and this dust and moisture is churned into the lubricant. As this progresses, the material becomes like putty. The most effective way to prevent this is to keep the gear case full of lubricant at all times. Put enough lubricant in so that some is flushed out at the top. Then at the proper intervals, as determined by operation of the particular machine and gear case involved, clean out the gear case, flush and completely refill with a new change of lubricant. (Wood)

Q. What is viscosity and why use SAE numbers for their identification?

A. Viscosity is indicated in terms of Saybolt Seconds at 100 or 210 F. It is a measure of the fluidity of an oil. SAE numbers are indicative of viscosity ranges according to internal combustion engine requirements. Charts are available which show these ranges. Due to the relation to

automotive service, we list both the SAE number and the Saybolt Seconds Universal. Neither have any relation to quality of an oil. (Slomer)

Q. Can an over-all unit cost of mine machinery lubrication per year per thousand tons of coal be stated?

A. I know coal companies that figure unit cost of labor and supplies, but they don't release this information for publication. I have seen the maintenance cost of certain cutting machines cut from $28.00 per thousand tons of coal to $8.00 per thousand, simply by revising the design of the lubrication system and training the operators of the machines in the proper method of application. This same organization made a study of mica ring failures, as directly attributed to over-lubrication. The frequency of lubrication was decreased on the commutator end bearing and the mica ring failures were eliminated. (Robert)

A. As an operator, I might add that the cost per ton for lubrication is variable, depending on the coal seam and conditions. In Pennsylvania, the cost of lubricant alone will probably average around one cent a ton, maybe a little more, maybe a little less in some cases where they pay strict attention to handling and application. The cost of lubricant is so small as compared with the cost of broken down machinery that the principal thing that we are concerned with is protecting the lubricant against contamination, getting it in the right place and trying to prevent breakdowns while the machine is in service. (Wood)

A. To Mr. Wood's comments I would like to add lubricant cut figures for various operations. For example, in conveyor mines a quarter of a cent a ton would be average, in a mobile mine maybe a cent a ton. For mobile mines with large pieces of mining equipment, the cost of lubricants might run as high as two and a half cents a ton. In other words, there is quite a variation in operations, depending on the type of equipment in use. (Robert)

(Men of Lubrication, from p. 11)

nature of their endeavors. The work of A. Fogg on parallel thrust surfaces with his theory of the so-called "thermal wedge" is an important addition to our understanding of a significant secondary effect. The application of hydrostatic lubrication to various problems is also an important development. (This principle is not new. A paper in **Engineering,** December 14, 18, 1917, "A Simple Problem in Forced Lubrication" by Lord Rayleigh gives the results for a footstep bearing lubricated in this way and appears to be the first paper on this subject.)

The effect of varying loads and speeds in a film lubricated journal bearing poses some difficult problems. Some progress has been made here by J. T. Burwell and H. W. Swift and others but much remains to be done in the field of dynamic loading.

Solutions to the mathematical problems arising when the side leakage in a bearing is considered have been few. Until quite recently Michell's solution to the inclined slider bearing was the only one that had been obtained. Within the last few years new solutions have been given for the slider bearing problem by A. Charnes and E. Saibel and also F. Archibald. The side leakage problem in the usual journal bearing problem has proved intractable though a solution has been obtained for a very special kind of bearing.

The publication by M. C. Shaw and E. F. Macks of their comprehensive book "Analysis and Lubrication of Bearings", and A. G. M. Michell's more recent book "Lubrication" is a valuable addition to the literature of the subject in the recent post-war period.

ACKNOWLEDGEMENT: It is impractical to credit all the sources of information used but special mention should be made of the help and encouragement given the author by Mayo D. Hersey.

(Lube Lines, from p. 7)

from the United States. As a consequence, virtually all the expansion in productive capacity which has been installed in the last few years, or is nearing completion, has been for HVI solvent quality. However, in view of the large number of different specifications of lubricants which are required in the world economy there is no doubt that the rest of the world will find it desirable to continue to import certain grades from the U. S. One such example is Pennsylvania Cylinder Oils.

The use of chemical additives for improving the quality and useful life of lubricants has been expanding rapidly. While much of the early development work was carried out in the United States, there has been considerable activity in this direction for some time now in Western Europe. As a result, the manufacture of such additives has now begun and should continue to increase. This will not only make areas outside the United States less dependent on us for such additives, but will undoubtedly tend to accelerate the use of aditive-type lubricating oils world-wide.

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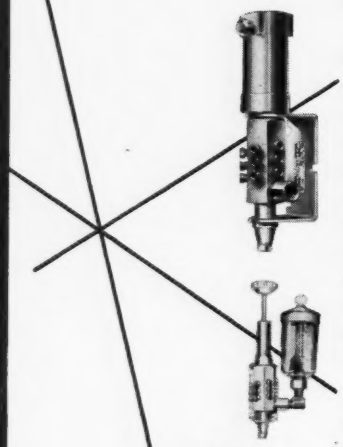

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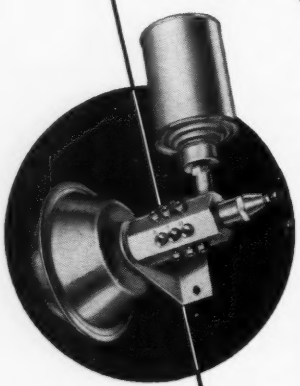

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PIONEER BUILDERS OF LUBRICATING EQUIPMENT FOR A QUARTER CENTURY

(Personals, from p. 34)

H. M. Fraser, International Lubricant Corp., New Orleans, La., has been awarded the '54 NLGI Award for Distinguished Achievement for his development of hydroxy-stearate soaps as a base for a single grease that would fill the multi-purposes of automotive and industrial needs.

E. J. Gesdorf has been named Senior Application Engineer of The Farval Corp., Cleveland, Ohio.

Obituaries: **F. R. Truby** (ASLE Philadelphia Section), Armstrong Cork Co., Lancaster, Pa., Aug. 11, 1954; **Dr. R. H. Patch,** Vice-President — Operations, E. F. Houghton & Co., Hot Springs, Ark., Nov. 13; **F. L. Binford,** President of D-A Lubricant Co., Indianapolis, Ind., Nov. 16.

(Current Literature, from p. 36)

or more experts. Illustrated with 37 pictures, the manual takes the serviceman step by step through the fifty-five parts of the procedure that is now the official recommendation of the NLGI. (NLGI, 4638 J. C. Nichols Pkwy., Kansas City, Mo., 15c ea.)

OILLESS BEARINGS. Reprints (No. 164) are now available of "Oilless Bearings — Their Place in Industry," a paper presented by C. D. Spadone before the Association of Iron & Steel Engineers. (Spadone-Alfa Corp., Box 523, South Norwalk, Conn.)

SEPARATORS. A new catalog, "Colloidair Systems for Recovery of Suspended Materials, for Waste Water Treatment, Etc.," has been released featuring factual data on the efficiency of the Colloidair process, flow sheets of typical installations, description of operating principles, and an analysis by types of industries of various applications for Colloidair systems. (Bulkley, Dunton Processes, Inc., 295 Madison Ave., New York 17, N. Y.)

PIPE-COUPLING COMPOUND. Bulletin No. 26-A describes a new theory, coincident with the introduction of a new

Liqui-Moly product called High Pressure Silicone Thread Compound, of making pipe-joints pressure-tight by lubricating the joint as it is being made up so well that, with little extra torque, the fitting can be run up two or three turns past the end of the thread producing a perfect seal. (The Lockrey Co., Southampton, N. Y.)

WAY LUBRICANT. Sunoco way lubricant, especially compounded to eliminate stick-slip or "jumpy table" on the ways of machine tools, is fully described in a new Sunoco Technical Bulletin. (Sun Oil Co., 1608 Walnut St., Philadelphia 3, Pa.)

(New Products, from p. 28)

rex glass reservoirs, metal parts are of machined brass with the base of durable cast aluminum. (Trico Fuse Mfg. Co., 2948 N. 5th St., Milwaukee 12, Wisc.)

THREAD LUBRICANT. A new thread lubricant for use on titanium, stainless steel, and aluminum has been announced that is available in easy-to-use tubes in two grades: Blue Goop for temperatures up to 400 F. (eliminates galling so threads can be reused over and over again), and Silver Goop for temperatures up to 2,100 F. (an effective antiseize compound for use on threaded parts of stainless steel and high temperature alloys). (Crawford Fitting Co., 884 E. 140th St., Cleveland 10, Ohio.)

AUTOMATIC TIME & CONTROL. Automatic application of fluid lubricants to bearings on individual industrial machines at predetermined intervals is provided by a new air-operated centralized lubrication system. Developed to fulfill the need for servicing machines operating at varying rates of speed where applications of lubricant at varying intervals may be required, control may be by any one of three methods: (1) Mechanical — utilizing motion of the machine to actuate an air valve which in turn cycles the lubrication system; (2) Elec-

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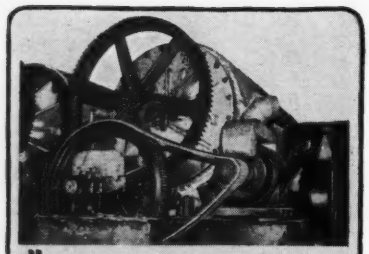

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trical — utilizing an adjustable time clock to actuate a solenoid air valve which in turn cycles the lubrication system at intervals of 7½ minutes to 3 hours; and (3) Manual — utilizing an air valve actuated at machine operator's discretion. (Brochure 806, Lincoln Engineering Co., 5743 Natural Bridge Ave., St. Louis 20, Mo.)

ANTI-RUST SPRAY. Gard, a new rust proofer containing silicones and polar compounds specially formulated to prevent rust and corrosion in outdoor storage for a year, and in indoor storage for five years, is completely compatible with all lubricating oils and greases. It can be polished to provide an almost invisible protective film, or it can be left intact as sprayed giving a slightly more durable and visible protective film; if applied directly to wet metal, it will completely displace all water present. (Gard Industries, Inc., 733 Green Bay Rd., Wilmette, Ill.)

VARIABLE SPEED DRIVES. Recently announced as the newest addition to its line of power transmission equipment, the Cleveland Speed Variator is available in 9 sizes ranging from one-half to 10 hp. at 1750 input rpm. Power is transmitted from the input shaft to the output shaft through alloy steel driving balls which are in pressure contact with discs attached to the two shafts; relative speeds of the two shafts are adjusted through a 9:1 range (from 1/3 to 3 times the input speed) by changing the angular positioning of the axles on which the balls rotate. Speed is regulated by a simple manual adjustment, with indicator, or by manual or power operated remote control devices. (Bulletin K-100, The Cleveland Worm & Gear Co., 3249 E. 80th St., Cleveland 4, Ohio.)

(Section News, from p. 21)

Electric; **March** — "Filtration & Reclamation of Industrial Oils," by J. R. McCoy, Honan-Crane Corp., & R. C. Fatout, Alvord Oil Co.; **April**—"Care & Maintenance of Hydraulic Systems," Vickers Inc.; **May**—"Gear Lubes & Anti-Friction Bearings," Standard Oil Co. (Ind.); **June**—"Plant Lubrication Program." (Submitted by R. C. Fatout, Sec'y-Treas.)

KINGSPORT, December '54. J. F. Collins, The Texas Co., presented a paper entitled "Laboratory Analyses of Lubricants with Correlation of Such Analyses to Field Practice." (Submitted by J. E. Fleenor, Sec'y-Treas.)

MILWAUKEE, November '54. R. G. Tiffany, Dow-Corning Co., presented a paper entitled "What is a Silicone?," followed by a film on "High Temperature Lubricants."

December. Annual Christmas Party and Casino Nite. (Submitted by J. A. Gramling, Sec'y.)

MONTREAL, October '54. R. Q. Sharpe, Socony-Vacuum Oil Co., Inc., presented a paper entitled "Hydraulic Fluids—Their Selec-

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tion & Maintenance." (Submitted by J. H. C. Lemeilleur, Sec'y.)

NEW YORK, September '54. Panel session on "Today's Automobile," with O. W. Wuerz, Cab Service & Parts Corp., as Moderator; Panelists included: H. R. Wolfe, Research Laboratory Div., GMC; C. M. Heinen, Chrysler Corp.; R. S. Gasvoda, Ford Motor Car Co.; and J. T. Bugbee, The Texas Co.

October. C. W. Nichols, Jr., Socony-Vacuum Oil Co., Inc., presented a paper dealing with normal laboratory tests used in evaluting lubricating oils entitled "What Happened To My Oil?"

November. John Boyd (ASLE President), Westinghouse Electric Corp., spoke on the subject of "The Influence of ASLE on Present Industry Practices." (Submitted by E. Landau, Section Director.)

N. CALIFORNIA, November '54. H. L. Memmingway (President-Elect of N.L.G.I.), The Pure Oil Co., presented a paper entitled "More Power To You." " . . . The past five or six years represent one of the greatest periods of advancement and change in automotive engine design and probably the greatest increase in average engine power output of any period in automotive history. The unprecedented acceptance of automatic transmissions and other automotive accessories, plus the trend toward V-8 engines of higher compression ratio and greater power, poses new problems in the development of suitable lubricants and fuels for these engines . . ." Mr. Hemmingway discussed why these problems have developed and of the means being used by both the engine manufacturer and the refiner to solve these problems.

December. M. K. Gebert, The National Motor Bearing Co., presented a paper entitled "Shaft-Type Oil Seals," reviewing current oil seal practice and outlining the anticipated future demands of industry, and the relationship between the oil seal and lubrication industries.

June 20 thru July 1, 1955. A

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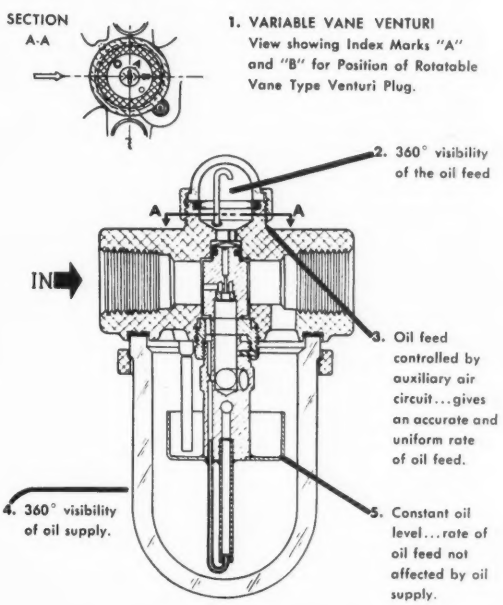
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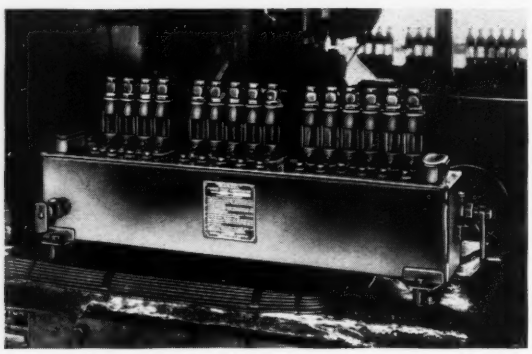

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two-week summer course in "Lubrication Engineering" co-sponsored by the University of California Engineering Extension and the ASLE Northern California Section. For further details, see "Lube Lines." (Submitted by G. H. Hommer, Section Reporter, & A. C. West, Sec'y-Treas.)

ONTARIO, November '54. Election of Officers (see ASLE Directory. Submitted by I. E. McIntosh, Sec'y.)

TWIN CITIES, December '54. E. T. Neubauer, Trane Co., presented a paper covering oil specifications, bearing surface finishes, and lubricant deterioration, entitled "Lubrication & Specifications." (Submitted by C. D. Johnson, Sec'y.)

(Patent Abstracts, from p. 39)

pentasulfide, and elemental sulfur at a temperature range of from about 115 C. to about 130 C. for a period of time not substantially exceeding four hours, the reactants being employed in the proportion of at least about 4 mols of the bicyclic terpene and from about 0.5 to about 3 mols of sulfur for each mol of phosphorous pentasulfide.

Grease Compositions Containing an Aryl Oxy Alkyl Salt as a Stabilizer, Patent No. 2,690,429, by A. J. Morway & P. V. Smith, Jr., assignors to Standard Oil Development Co.

A lubricating grease composition consisting essentially of a lubricating oil thickened to grease consistency with from 5% to 20% by weight of a mixture of from one to three parts of an alkali metal soap of a high molecular weight fatty acid and from one to two parts of an alkali metal salt.

Sulfurized Sperm Oil Lubricating Oil Additives Stabilized with Sulfites, Patent No. 2,690,445, by E. J. Corcoran & E. P. Cashman, assignors to Standard Oil Development Co.

A process for the stabilization of the sulfur content of lubricating oil additives of the class consisting of sulfurized and sulfurized-phosphorized sperm oil which comprises admixing with said additive a material of the class consisting of sodium sulfite and sodium hydrosulfite, heating the mixture to a temperature within a range of from 150 to 250 F. for from 5 minutes to one hour, and separating the improved additive from the mixture.

Metal Soap Grease Containing Alkaline Earth Metal Phenolate, Patent No. 2,690,998, by J. P. Dilworth, C. H. Culnane & H. V. Ashburn, deceased,

assignors to The Texas Co.

A lubricating grease composition comprising as the essential constituents an oleaginous liquid lubricating base as the major constituent, about 10-30% by weight of a lithium soap of a saturated soap-forming fatty material, said grease normally being corrosive to copper, and from 0.25 to 5% by weight based on said composition of an oil-soluble alkaline earth metal salt of an alkyl phenol having a total of 10 to 30 carbon atoms in the alkyl substituents on each benzene nucleus, said salt being in sufficient proportion to enable said grease composition to pass the copper corrosion test of U. S. Army Specification 2-134.

Silver Protective Agents for Sulfur-Containing Lubricants, Patent No. 2,690,999, by W. Lowe & J. O. Clayton, assignors to California Research Corp.

A lubricant consisting essentially of a major portion of a sulfur-containing lubricating oil composition severely corrosive to silver, and a small amount, sufficient substantially to reduce the sulfur attack on silver, of a 1,3,4-thiadiazoly 1-2,5-bis(dialkyl dithiocarbamate), said alkyl radical containing from 1 to 20 carbon atoms.

Lubricating Oils, Patent No. 2,691,000, by J. S. Elliott, assignor to C. C. Wakefield & Co., Ltd.

A lubricating composition comprising a mineral oil containing a small amount sufficient to increase the load-carrying capacity of the oil of an ester of acetic acid substituted in the alpha position by an element selected from the group consisting of sulphur and selenium.

Mineral Lubricating Oil Composition, Patent No. 2,691,001, by E. B. Cyphers & D. W. Young, assignors to Standard Oil Development Co.

A mineral lubricating oil composition containing a major proportion of a mineral lubricating oil and dissolved therein a minor oxidation inhibiting amount of 4,4'-isopropylidine bis (2-isopropylphenol).

Method of Processing Solvent-Extracted Lubricating Oil by Treatment with Phosphorus Pentasulfide and Potassium Hydroxide and the Resulting Products, Patent No. 2,691,002, by J. D. Bartleson & H. E. Alford, assignors to The Standard Oil Co.

A method of processing solvent-extracted lubricating oil stock consisting essentially of hydrocarbon material to yield an oil having improved properties in service which comprises treating said stock with an amount of phosphorus pentasulfide in the range of about 0.1 to about 0.75 by weight, at a temperature in the range of about 275 to 450 F., then with potassium hydroxide in an amount more than eight equivalents per mol of the phosphorus pentasulfide at a temperature of at least 325 F.

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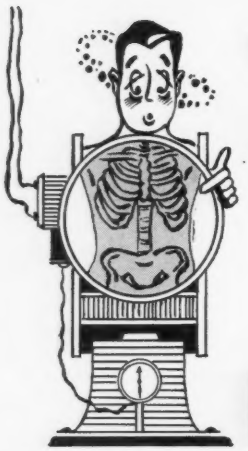

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FIG. 811

This vane-type Bowser Indicator is spring-actuated, thus providing an indication of the quantity flowing in the line.

FIG. 811

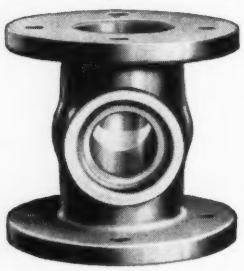

FIG. 54A

This Bowser Flow Sight is recommended for use on gravity lines where the flow is vertically downward.

FIG. 54A

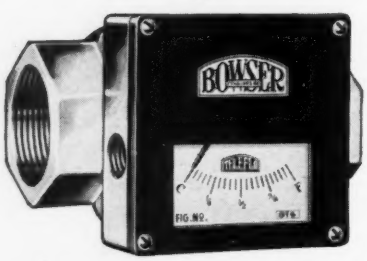

FIG. 816

The Figure 816 Bowser Teleflo Indicator is equipped to operate a gong or to stop pump motor if liquid flow stops.

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FIG. 816

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Lubricants Comprising Waxy Mineral Oil, Silicates, and a Phthalyl Chloride Acylated Tetra Paraffin-Alkylated Phenol, Patent No. 2,691,633, by G. J. Benoit, Jr., assignor to California Research Corp.

A lubricant composition comprising a mixture of a waxy mineral oil and a tetra-(branched-chain alkyl) silicate, together with from about 1 to 75% by volume of the mineral oil and from 75 to 25% by volume of the silicate together with from about 1 to 3% by weight of a phthalyl chloride acylated tetraparaffin-alkylated phenol.

Lubricant Composition Comprising a Mixture of Waxy Mineral Oil, Poly-1,2 - Oxy-Propyleneglycol and a Phthalyl Chloride Acylated Tetraparaffin-Alkylated Phenol, Patent No. 2,691,634, by G. J. Benoit, Jr., assignor to California Research Corp.

A lubricant composition comprising a mixture of a waxy mineral oil and a poly-1,2-oxypropylene glycol in the proportion of from 25 to 75% by volume of the mineral oil and from 75 to 25% by volume of the glycol, together with from about 2% to about 3%, by weight, of a phthalyl chloride acylated tetraparaffin-alkylated phenol.

Copolymers of Multiolefins with Vinyl Ethers and Method of Making Same, Patent No. 2,691,646, by D. W. Young & W. J. Sparks, assignors to Standard Oil Development Co.

A process for the preparation of lubricating oil additives which comprises copolymerizing from about 90% to 30% of the vinyl ether of a mixture of alcohols having an average number of carbon atoms of about 13.5 obtained by the hydrogenation of coconut oil with from about 10% to about 70% by weight of a conjugated hydrocarbon of a total of from 5 to 6 carbon atoms having 4 to 5 carbon atoms in a straight chain and having a methyl group in the 2-position at a temperature of about -25 C. in the presence of a methyl chloride solution of $AlCl_3$ for a period of time sufficient to give an oil soluble copolymer of a molecular weight of about 5,000 to 30,000.

Lubricating Grease Compositions from Hydroxy Phenyl Fatty Acids, Patent No. 2,692,232, by J. H. Bartlett, A. J. Morway, & G. E. Serniuk, assignors to Standard Oil Development Co.

A lubricating grease composition comprising a lubricating oil thickened to a grease consistency with an alkali metal soap of a hydroxy phenyl fatty acid containing about 18 carbon atoms in the non-aromatic portion thereof.

Synthetic Lubricant, Patent No. 2,692,233, by D. L. Cottle, D. W. Young & F. Knoth, Jr., assignors to Standard Oil Development Co.

A synthetic lubricating oil having outstanding lubricating characteristics at low temperatures which consists essentially of a mixture of diesters of adipic acid.

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PRINTED IN U.S.A.